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INVESTIGATION OF GAS LIQUEFIERS FOR SPACE OPERATION

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FOREWORD

This report was prepared by Malaker Laboratories, Inc., under USAF Contract No. AF 33(657)-10,133. This contract was initiated under Project No. 8169, Task No. 816903, "An Analytical Investigation of Gas Liquefiers for Space Operations." The work was administered under the direction of the AF Aero Propulsion Laboratory, Aeronautical Systems Division, with Mr. Charles W. Elrod acting as Project Engineer.

This report covers work conducted from October 1962 to May 1963.

ABSTRACT

A detailed description and comparison is given of thermodynamic cycles suitable for use in space for the liquefaction and reliquefaction of He, H_2, N_2, F_2 and O_2 at rates from 1 lb/day to 1000 lbs/day. A survey is presented of the mode of operation, performance and efficiency of the following cycles: cascaded compressed vapor cycles, systems using Joule-Thomson ~~(J-T)~~ cooling only, systems using expansion engines, Stirling cycle systems, Taconis cycle systems and miscellaneous systems. Tables and graphs are included presenting the significant data on performance of each suitable cycle based on conservative estimates of the current state-of-the-art and on optimistic estimates of the probable future state for the period 1965-1970.

A survey is given of space environment and detailed discussions are presented on the question of component compatibility and design for space operation, particularly covering such matters as component weights, volumes and reliability.

Definitive conclusions are arrived at which serve to permit specific recommendations for future action and a summary is made of some current areas of ignorance and of the more significant problem areas which are discussed in detail in the main body of the report.

TABLE OF CONTENTS

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

TABLE OF CONTENTS (Cont'd)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Cont'd)

LIST OF ILLUSTRATIONS (Cont'd)

LIST OF ILLUSTRATIONS (Cont'd)

LIST OF ILLUSTRATIONS (Cont'd)

LIST OF ILLUSTRATIONS (Cont'd)

LIST OF ILLUSTRATIONS (Cont'd)

LIST OF TABLES

LIST OF TABLES (Cont'd)

LIST OF TABLES (Cont'd)

LIST OF TABLES (Cont'd)

I. INTRODUCTION

This investigation has as its objective the determination of the thermodynamic cycles best suited for use in space for the liquefaction and reliquefaction of helium, hydrogen, nitrogen, fluorine and oxygen at rates from 1 lb/day to 1000 lbs/day.

The report is subdivided into nine sections. In Section II some data on the subject gases are presented along with references to fuller data. Section III defines the ranges of liquefaction and reliquefaction rates covered by this report. Section IV gives a theoretical survey of the relative efficiencies of liquefaction and reliquefaction for ideal reversible cycles. Section V gives a detailed survey of actual thermodynamic cycles used for refrigeration and liquefaction and covers all well known cycles, whether suitable or unsuitable for space operation. The cycles covered include: cascaded compressed vapor cycles, systems using Joule-Thomson cooling only, systems using expansion engines, Stirling cycle systems, Taconis cycle systems and miscellaneous systems. The treatment is full and charts and graphs are included presenting the significant data on performance of each suitable cycle based on conservative estimates of the current state of the art and on optimistic estimates of the probable future state for the period 1965-1970. Although none of the miscellaneous systems described in this report appear likely to be suitable for the subject task in the period 1965-1970, descriptions (in some cases quite detailed descriptions) of these have been included for the sake of completeness and future reference.

Section VI presents some data on compressors, motors and expanders, including estimates of weights and volumes. Section VII surveys the space environment, including space temperatures, and discusses component compatibility and design. In this discussion data are presented for estimating radiator weights and areas. Section VIII gives a detailed qualitative and quantitative comparison of all promising cycles, including data on performance, weights, volumes, reliability, compatibilites with space environment, etc. This comparison is for all sizes of cycles from miniature to those for production up to 1000 lbs/day of the subject gases and is supported by numerous charts, tables and graphs. Definitive conclusions are arrived at which serve to permit specific recommendations for future action to be made in Section IX. In addition Section VIII contains summaries of some of the areas of ignorance and of the more significant problem areas and possible future breakthroughs which have been discussed in the main body of the report.

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II. DATA ON SUBJECT GASES

This investigation concerns the liquefaction and reliquefaction in space environment of helium, hydrogen, nitrogen, fluorine and oxygen.

Tables 1 and 2 give pertinent data concerning the subject gases. Similar data is also included in these tables for neon. Although neon is not one of the subject gases of the investigation, data concerning it are included since it may be suitable for use for the refrigeration and/or reliquefaction of the subject gases.

For detailed properties of the subject gases, such as the presented in S-T diagrams, etc., reference is made to the publications listed below under "Bibliography for Thermodynamic Data." Since these references are readily available, the charts contained therein are not duplicated in this report.

TABLE 1

PROPERTIES OF LIQUEFIED GASES

Fluid	Normal Boiling Point		Critical Temp.		Critical Press. Atm.	Inversion Temp.	
	°R	°K	°R	°K		°R	°K
Hydrogen	36.7	20.4	59.7	33.2	12.98	368	204.6
He^4	7.6	4.2	9.33	5.19	2.26	~90.9	~50.5
Nitrogen	139	77.3	226	126.0	33.5	1120	621
Oxygen	162	90.1	278	154.3	49.7	1609	893
Flucrine	153	85.0	259	144.1	55	-	-
Neon	49.0	27.2	79.7	44.4	25.9	~396	~220

TABLE 2

SPECIFIC GRAVITY AND LATENT HEAT OF CRYOGENIC LIQUIDS AT NORMAL BOILING POINT

Substance	d	L (joules/g)
H_2	0.071 gm/cm^3	452
He^4	0.122	20.4
N_2	0.808	199
O_2	1.14	213
F_2	1.50	172
Ne	1.205	89.4

BIBLIOGRAPHY FOR THERMODYNAMIC DATA

General Data:

1. Johnson, V. J. (editor): A Compendium of the Properties of Materials at Low Temperature (Phase I); WADD Technical Report 60-56, Part I; July 1960. This report contains charts and tables of the following properties: Density, Expansivity, Thermal Conductivity, Specific Heat and Enthalpy, Transition Heats, Phase Equilibria, Adsorption, Surface Tension and Viscosity on the following fluids: Helium, Hydrogen, Neon, Nitrogen, Oxygen, Air, Carbon Monoxide, Fluorine, Argon, and Methane.

2. Chelton, D. B. and Mann, D. B.: Cryogenic Data Book; WADC Tech. Report 59-8; March 1959. This report contains a compilation of the physical properties of the following liquefied gases: helium, hydrogen (para and ortho), hydrogen deuteride, deuterium, nitrogen, oxygen, and air.

Nitrogen Data:

1. Keesom, W. H. and Houthoff D. J.: Leiden Comm. Suppl. 65c (1928).

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2. Keesom, W. H. and Houthoff, D. J.: Leiden Comm. Suppl. 65d (1928).

He^4 Data:

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Fluorine Data:

1. General Chemical Division, Allied Chemical Co.: Technical Bulletin TA-85411 (1961).

2. Hu, J. H., White, D. and Johnston, H. L.: ASTIA report No. AD 10067 (1953).

Neon Data:

1. National Bureau of Standards, Cryogenic Engineering Laboratory (1961) (Gas-Phase only).

2. Troyer, B. D. and Timmerhous, K. D.: Advances in Cryogenic Engineering, Vol. 6 (1961).

III. RANGES OF LIQUEFACTION AND RELIQUEFACTION RATES.

The ranges of liquefaction and reliquefaction rates covered in this study are from 1 to 1000 lbs/day. Table 3 shows these ranges expressed not only in lbs/day, but also in liters liquid/hour and in watts (refrigeration for reliquefaction) for the subject gases. In addition, Figure 1 shows the refrigeration load for reliquefaction in watt-hours/lb plotted against the storage pressure for the subject gases.

TABLE 3

REFRIGERATION REQUIRED FOR RELIQUEFACTION

Substance	Boiling Point at 1 Atmosphere °K	Boiling Point at 1 Atmosphere °R	lbs/day	liter/hr.	Watts Refrigeration For Reliquefaction
He^4	4.2	7.6	1-1000	0.155-155	0.107-107
H_2	20.4	36.8	1-1000	0.266-266	2.37-2,370
N_2	77.3	139	1-1000	0.023-23.4	1.05-1,050
O_2	90.1	162	1-1000	0.0166-16.6	1.12-1,120
F_2	86.1	155	1-1000	0.0126-12.6	0.90-900

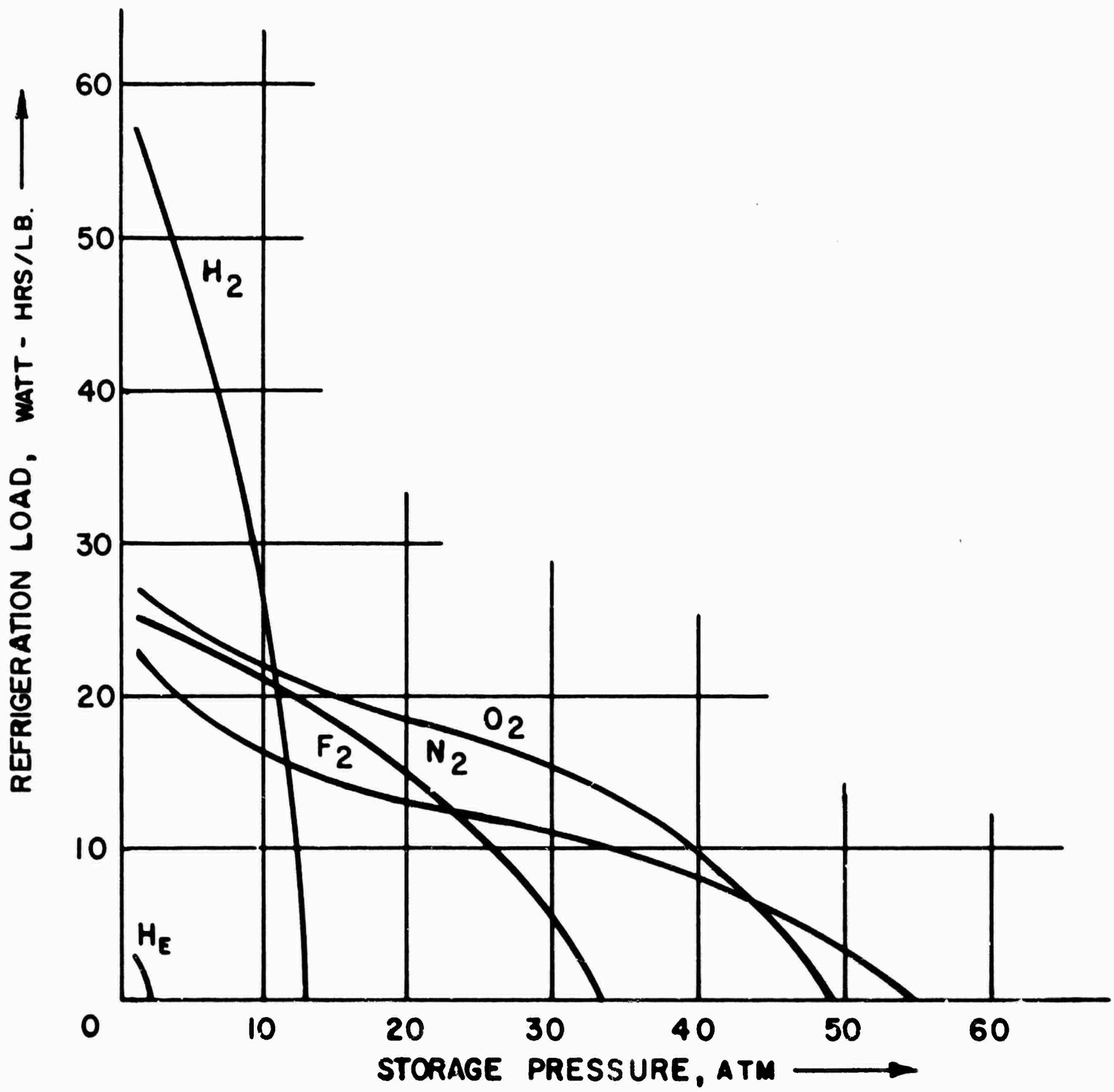

Fig. 1 - Refrigeration load for reliquefaction of subject gases as a function of storage pressure.

IV. RELATIVE EFFICIENCIES OF LIQUEFACTION AND RELIQUEFACTION.

In this report consideration will be given to both liquefaction and reliquefaction of the subject gases. In reliquefaction it is assumed that by suitable placement of a refrigerating device the boil-off gas can be reliquefied ideally at the temperature and pressure at which it is stored. In the reliquefaction process, therefore, it is not necessary to expend energy in cooling the gas down from ambient temperature to the temperature of the boiling liquid and in consequence the refrigerating device is not loaded by the sensible heat of the gas to be reliquefied. One would expect, therefore, that the energy required for reliquefaction would be smaller than for liquefaction and the following section presents a theoretical assessment of these differences for ideal liquefiers and ideal reliquefiers.

Before considering some of the relative merits of different liquefaction and reliquefaction techniques, it is of interest to assess the theoretical maximum efficiencies of both processes.

First consider an idealized refrigerator in which the working gas is first isothermally compressed from an initial state 1 at pressure p_1, temperature T_1 and entropy S_1 per mole to a pressure p_2, and entropy S_2 per mole. The pressure p_2 is so chosen that a subsequent reversible isentropic expansion of the gas to the pressure p_1 will result in its being completely liquefied. Let this final completely liquid state, 3, be at a temperature T_3 and entropy S_3 per mole.

The work of compression, W_c, per mole then is given by:

$$W_c = T_1 \, (S_1 - S_2) \tag{1}$$

but since $S_2 = S_3$ by definition of an isentropic expansion we have

$$W_c = T_1 \, (S_1 - S_3) \tag{2}$$

To complete the refrigerative cycle, let the liquid evaporate completely at pressure p_1 and temperature T_3 and then subsequently warm as a vapor from T_3 to the initial temperature T_1, also at pressure p_1, the heat required for this coming from the surroundings. It can readily be shown* that the net work done on one mole of gas in the cycle is:

$$W = T_1 \, (S_1 - S_3) - (H_1 - H_3) \tag{3}$$

where the H is the enthalpy per mole at the points 1 and 3.

*Daunt, J. G. Hdb d. Physik, 14.1. (1956)

By using such a refrigerator to liquefy the working fluid it is found that the minimum work of liquefaction, W_{min}, is also given by equation (3). Table 4 lists some values of W_{min} obtained in this way from known entropy and enthalpy tables.

To calculate the minimum work required for reliquefaction, one can assume that the refrigerator is ideal with efficiency equal to that of a reversible thermodynamic engine operating between the ambient or sink temperature, T_1, and the boiling point, T_3, of the gas to be reliquefied. Under this assumption the minimum work is

$$W_{min} = L\left(\frac{T_1}{T_3} - 1\right) \tag{4}$$

where L is the latent heat of evaporation of the subject gas.*

Table 5 sets out the evaluations of this theoretical minimum work of reliquefaction. It will be seen that it is indeed less than the theoretical minimum work for liquefaction and Table 6 gives the ratio of the minimum work for liquefaction to the minimum work for reliquefaction.

In practice, for a variety of reasons the work of liquefaction and reliquefaction greatly exceeds these minimum figures. The ideal processes cannot be realized in practice, because of the irreversibilities inherent in practical processes.

TABLE 4

MINIMUM WORK REQUIRED FOR LIQUEFACTION

Substance	p_1 (atm.)	T_{comp} (°K)	W_{min} (KW-hr/liter liquid)
Oxygen	1	293	0.205
Nitrogen	1	293	0.169
Hydrogen (Normal)	1	293	0.224
Helium (He^4)	1	293	0.235

*If the refrigerative load is not isothermal then, as shown by Jacobs, R. B. (Adv. in Cryogenic Eng. 7, 567, 1962), the low temperature in the expression for the efficiency of a Carnot refrigerator must be replaced by the log-mean temperature.

TABLE 5

MINIMUM WORK REQUIRED FOR RELIQUEFACTION

Substance	P* (atm.)	T** (°K)	W_{min} (KW-hr/liter liquid)
Oxygen	1	293	0.159
Nitrogen	1	293	0.127
Hydrogen (Normal)	1	293	0.119
Helium (He^4)	1	293	0.049

*Pressure of evaporating liquid
**Temperature of refrigerator heat sink

TABLE 6

RATIO, r, OF MINIMUM WORK FOR LIQUEFACTION TO MINIMUM WORK FOR RELIQUEFACTION (PRESSURE OF LIQUID = 1 ATM; TEMPERATURE OF HEAT SINK = 293°K)

Substance	r
Oxygen	1.69
Nitrogen	1.38
Hydrogen (Normal)	1.88
Helium (He^4)	4.75

Irreversibilities are always introduced in heat flow processes, for example, in all heat exchangers, evaporators and condensers in which heat flow occurs always under a temperature gradient. It is the purpose of the main body of this report to assess the irreversibilities that are introduced in the various liquefaction and reliquefaction systems under consideration.

V. SYSTEMS FOR LIQUEFACTION AND RELIQUEFACTION

V.0 General References

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0.17 Gardner, J. B. and Smith, K. C.: "Power Consumption and Thermodynamic Reversibility in Low Temperature Refrigeration and Separation Processes," in Advances in Cryogenic Engineering, Vol. 3, Plenum Press, 1960.

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V.1. CASCADED COMPRESSED VAPOR SYSTEMS

1.1 INTRODUCTION

Compressed vapor refrigeration machines, with two notable exceptions, have not been used for the liquefaction or reliquefaction of the subject gases. The two notable exceptions are the air liquefier established by Kamerlingh-Onnes* and the nitrogen liquefier set up by Huguenin.** Although the systems are highly efficient, they require many working fluids and compressors and the associated mechanical complexity has in the past discouraged their widespread use. Further, the triple point of oxygen (54.4°K) seems to be a practical, if not theoretically absolute, lower limit to the temperature to which they can be operated. Because of these disadvantages it is unlikely that cascaded compressed vapor systems would be advantageous for use in space vehicles. On the other hand, compressed vapor refrigerators, used singly or in cascade, have been used quite extensively as precoolers in other refrigeration and liquefaction systems in order to increase the efficiency of the over-all systems. Moreover, their high efficiency renders them of some interest for possible O_2 and F_2 liquefaction. Because of these facts it seems appropriate to give a brief review of past and current capabilities of cascaded compressed vapor systems.

1.2 DESCRIPTION OF A SINGLE-STAGE CYCLE

Fig. 2 shows a flow diagram of a typical single stage compressed vapor refrigerator. It comprises a compressor which is used to liquefy the working fluid in the condenser, a condenser used to permit the fluid to be condensed, an expansion valve through which the compressed liquid expands into the evaporator and an evaporator which absorbs heat at a desired low temperature.

In practical single stage compressed vapor refrigerators the following requirements must be met: (a) the condenser temperature, T_2, must not be far above the critical temperature, T_c, of the fluid, (b) the pressure, P_2, of compression should not be too high, (c) the compression ratio, P_2/P_1, should be small, since the work of compression increases rapidly with increasing compression ratio, and (d) the low temperature, T_1, should be close to the boiling point, T_B, of the fluid in order that the inlet pressure to the compressor shall be approximately one atmosphere.

*Kamerlingh-Onnes, H. Leiden Comm. 14 (1894); Leiden Comm. Suppl. 35 (1913).

**Huguenin, A. Festschrift zum 70. Gerburtstag von Prof. A. Stodola. p. 272, Zurich 1929.

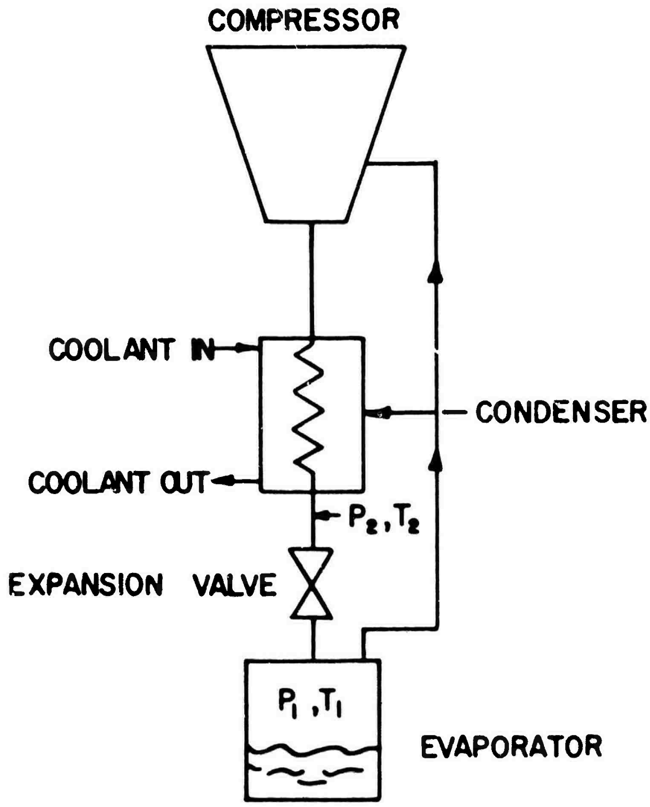

Fig. 2 - Flow diagram of single-stage compressed vapor refrigerator.

The conditions indicated above, together with practical considerations concerning, for example, the corrosiveness and/or toxicity of the working fluid, are sufficient to allow the choice of fluid required to operate between any two desired temperatures. Table 7 gives the necessary data on the properties of many low temperature working fluids for compressed vapor refrigerators.

TABLE 7

SOME DATA ON LOW TEMPERATURE FLUIDS FOR COMPRESSED VAPOR REFRIGERATORS

	Boiling Point °K	Critical Temp. °K	Critical Pressure atm.	Triple Point °K
Dichlorodifluoromethane (Freon 12)	243.1	384.2	39.6	118.3
Ammonia	239.6	405.5	111.5	195.5
Monochlorodifluoromethane (Freon 22)	232.5	369.1	48.7	113.5
Propane	230.5	368.7	44.0	85.0
Propylene	226.1	364.[illegible]	4[illegible].9	87.9
Carbon Dioxide		304.2	73.0	216.6
Monchlorotrifluoromethane (Freon 13)	191.7	302	39.4	92.2
Trifluoromethane*(Freon 23)	19[illegible]	[illegible]2.2	46.[illegible]	1[illegible]
Ethane	184.8	305.5	47.2	101.2
Nitrous Oxide	185.6	309.6	71.7	170.8
Ethylene	169.[illegible]	282.8	[illegible]	10[illegible].8
Tetrafluoromethane*(Freon 14)	[illegible]44.[illegible]	227.6	[illegible]6.7	82
Methane	111.7	190.6	45.8	90.5

*Data from Daunt, J. G., Hdb. d. Physik 14.1. (1956) and Kuprianoff J., Plank R. and Steinle H., Hdb. d. Kältetechnik Vol. IV. (1956).

1.3 DESCRIPTION OF CASCADED COMPRESSED VAPOR SYSTEMS

Cascading of compressed-vapor refrigerators can be used for going to temperatures down to that of liquid nitrogen or oxygen. It is theoretically possible to go below liquid oxygen temperatures by using liquid neon as the next refrigerant in the cascade. However, this requires reduced pressure in the oxygen evaporator and a high compression ratio in the neon circuit. To the authors' best knowledge, no cascade systems in practice have gone below the liquid oxygen-nitrogen range of temperatures.

In a cascaded system the evaporator of one refrigerator is used as the condenser for the next lower temperature refrigerator. In addition, the efficiency can be increased by inclusion of heat exchangers which allow exchange of heat between the gas evaporating from one evaporator and the incoming compressed gas to the next lower temperature refrigerator. In a fourfold cascaded system devised by Keesom* and subsequently put into practice by Huguenin**, ammonia, ethylene, methane and nitrogen were used as the working fluids, giving a practical efficiency of about 0.245 KW-hr per lb of N_2 liquefied.

Some analysis of the losses in Keesom's system has been given by Ruhemann ("The Separation of Gases," Oxford University Press, Second Edition, 1949, p. 139).

Modern commercial staged compressed vapor refrigerators are made by a variety of companies, and in general use the freon's as the working fluids. Two stage cascaded systems can easily reach to 165°K. Reference is made to work by Soumeral*** and Missimer**** for analyses of typical low temperature systems.

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*Keesom, W. H. Leiden Comm. Suppl. 76a. 1933

**Huguenin, A., Festschrift zum 70. Geburtstag von Prof. A. Stodola, p. 272, Zurich (1929).

***Soumeral, H. P., "Simplified Analysis of Two Stage Low Temperature Refrigeration Systems," Refrig. Eng. (July 1953), p. 746.

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V.2. SYSTEMS USING JOULE-THOMSON COOLING ONLY

2.1 SINGLE STAGE JOULE-THOMSON SYSTEMS WITHOUT PRECOOLING

2.11 RELIQUEFIERS OR REFRIGERATORS FOR TEMPERATURES ABOVE 77°K.

The simplest closed-cycle cooling system using Joule-Thomson expansion consists of a compressor, after-cooler, heat exchanger, and expansion valve. A system of this-type is shown schematically in Fig. 3.

The fluid enters the heat exchanger after compression to pressure p_1 and after cooling in the after-cooler (radiator) to the sink temperature, T_s, at the point a. It passes through the heat exchanger (path a-b), where it is cooled by gas from the evaporator. The fluid is then expanded isenthalpically at the valve (path b-c) to a pressure p_2 and liquid is produced. The pressure p_2 is the vapor pressure of the liquid at the evaporator temperature, T. This liquid is evaporated (path c-d) by the heat load, L, which is thermally connected to the evaporator and the cool gas, ideally at temperature T, is exhausted through the heat exchanger (path d-e). The gas at pressure p_2 emerging from the top of the heat exchanger at e is fed to the compressor, where it is compressed and returned to its original state at a, (p_1, T_s).

For an ideal system in which it may be assumed that there are no heat exchanger inefficiencies, no pressure drops in fluid flow, no heat leaks, etc., the thermodynamic cycle would be represented by the heavy curve in the T-S diagram of Fig. 4 in which the lettered points correspond to those of Fig. 3. For an ideally efficient cycle one would have:

$$T_s = T_e = T_a; \; T_c = T_d; \; p_1 = p_a = p_b; \; p_2 = p_c = p_d = p_e \qquad (5)$$

If h is the enthalpy in joules/g of the working fluid, then at the Joule-Thomson expansion at the valve $h_b = h_c$ and the heat exchanger heat balance is:

$$h_a - h_b = h_e - h_d$$

$$h_d - h_c = h_e - h_a \qquad (6)$$

But $(h_d - h_c) = (h_e - h_a)$ is the refrigerative capacity of the refrigerator at the low temperature per gram of working fluid passing through the cycle.

If n is the mass flow rate of the fluid in g/sec, then the refrigerative load, L, is $n(h_e - h_a)$ watts.

If it is assumed that the compression is isothermal at the sink temperature, T_s, ($T_s = T_e = T_a$, ideally), then the isothermal power of compression is

$$p_{iso} = n\ T_s\ (S_e - S_a)\ \text{watts} \tag{7}$$

where S is the entropy of the working fluid per gram. If the fluid on compression is treated as a perfect gas, then

$$p_{iso} = (nRT_s/M)\ \ln\ (p_1/p_2)\ \text{watts} \tag{8}$$

where M is the molecular weight of the working fluid and R the gas constant in joules/°K. The ideal coefficient of performance, C.P. (ideal), therefore is:

$$\text{C.P. (ideal)} = \frac{L}{p_{iso}} = \frac{(h_e - h_a)}{T_s(S_e - S_a)} \tag{9}$$

To allow for non-isothermal compression, for motor and motor drive inefficiencies and for the non-ideality of the cycle, let ξ be the isothermal compression efficiency, η the relative efficiency of the actual cycle and ζ the motor efficiency; then the actual coefficient of performance, C.P. (actual) is:

$$\text{C.P. (actual)} = \frac{L}{P} \approx \frac{M(h_e - h_a)}{RT_s\ \ln(p_1/p_2)}\ .\ \xi\eta\zeta \tag{10}$$

The thermodynamic cycle for such an actual refrigerator is shown in the T-S diagram of Fig. 4 by the broken curve.

2.12 COEFFICIENTS OF PERFORMANCE, POWER REQUIREMENTS FOR SINGLE STAGE J-T SYSTEMS FOR REFRIGERATION (RELIQUEFACTION) AT TEMPERATURES OF 77°K AND ABOVE.

The formula for the actual coefficient of performance, C.P. (actual), for single stage J-T cycle refrigerators operating down to 77°K has been given in Section 2.11 above. For this study in this temperature range we are concerned with the use of refrigerators for the reliquefaction of nitrogen, fluorine and oxygen. It is considered therefore that for operation of this kind of cycle, N_2 should be the preferred working gas, and the refrigerator would reliquefy N_2, F_2 and O_2 by condensation at the boiling points of these gases. It is possible, of course, that argon would be suitable as a working gas. However, since its boiling point is 87.4°K, it could only be used in a refrigerator for reliquefying O_2 unless the compressor inlet pressures were reduced below 1 atmosphere. This would result in great loss in the compressor volumetric efficiency and is in practice undesirable. Moreover,

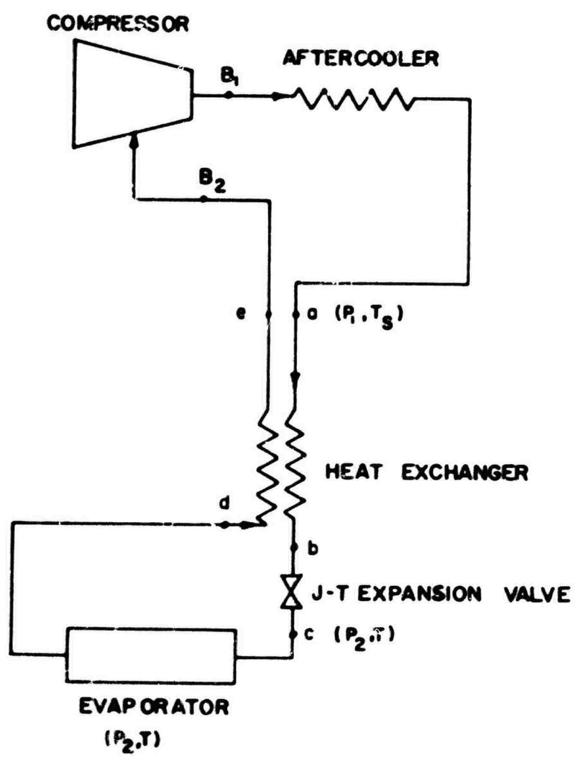

Fig. 3 - Flow diagram for single-stage Joule-Thomson refrigerator.

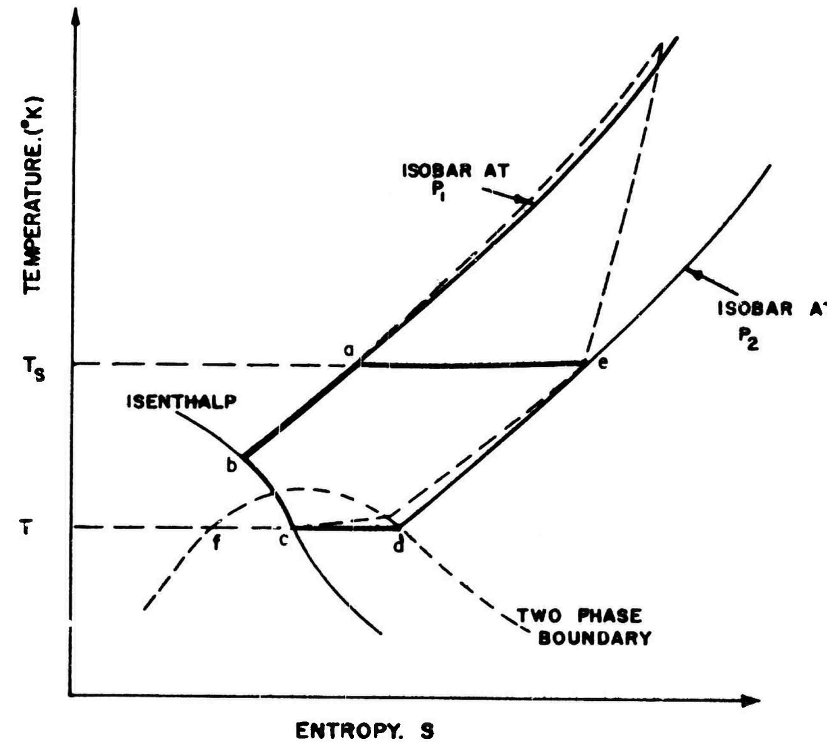

Fig. 4 - T-S diagram for single-stage Joule-Thomson refrigerator. (Full heavy lines are for ideally efficient cycle; broken lines are for actual cycle.)

being a monatomic gas the isothermal compressor efficiency would be less than that of N_2. For these reasons, therefore, we shall restrict our considerations to N_2 as the working fluid. (Clearly O_2 and F_2 would be undesirable for reasons of hazard and corrosion.) We shall be concerned therefore with the coefficients of performance of an N_2 system at 77°K, 85°K, and 90°K.

To allow subsequent exact comparison to be made with other cycles we shall adopt 300°K as the sink temperature, T_s. A discussion of the probable values of T_s is given in Section VII.2.51 and the effects of varying T_s are discussed in Sections VII.2.53 and V.4.4.

The compressor output pressure will be taken to be 2400 psia (approx. 163 atmos.), since previous work* has shown this to be a satisfactory operating pressure. If the pressure is reduced below this value, the coefficient of performance decreases markedly**, with consequent increase in power requirements, volume and weight of a system designed for a given refrigerative load. If one operates at higher pressures, serious practical difficulties arise in compressor design, compressor and system reliability, etc.

Table 8 presents the data for the coefficient of performance. It gives C.P. (Carnot) for an ideal Carnot refrigerator, where

$$C.P.\ (\text{Carnot}) = T/(T_s - T) \qquad (11)$$

It gives C.P. (ideal) for a single-stage J-T cycle with N_2 as working gas with input at 2400 psia and 300°K. It also gives C.P. (actual) for low power levels (100 lbs/day reliquefied) and for higher power levels (1000 lbs/day reliquefied). These evaluations were made assuming that ξ, the isothermal compressor efficiency, is 67% (see Section VI), that η, the relative efficiency of the actual cycle, is 85% for the higher power levels and 80% for the low power levels, and that ζ, the motor and motor drive efficiency, is 85% for the higher power levels and 80% for the low power levels. These figures for η are taken from the known performance of current systems employing this cycle*** and account for reasonable heat exchanger losses, heat leaks and other irreversibilities.

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*Daunt, J. G. Hdb. d. Physik 14, 1 (1956).
**See for example: Rousseau, J. Airesearch Mfg. Co., Dec. 1960, ASTIA Document No. 22204.
***See for example: Daunt, J. G. Hdb. d. Physik 14, 53 (1956).

TABLE 8

COEFFICIENTS OF PERFORMANCE FOR SINGLE-STAGE J-T CYCLE REFRIGERATORS USING N_2 AS WORKING FLUID (RELIQUEFIERS)

Temp, T °K	Reliquefied Gas	P2 Atmos.	C.P. (Carnot)	C.P. (Ideal)	C.P. (Actual) Low Power Level (100 lbs/day)	C.P. (Actual) Higher Power Level (1000 lbs/day)
77°K	N_2	1.0	0.347	0.061	0.026	0.029
85°K	F_2	2.4	0.396	0.071	0.031	0.035
90°K	O_2	3.6	0.423	0.077	0.033	0.037

The power requirements, P, in kilowatts for a given refrigerative load, L (in kW) is then given by:

$$C.P.\ (\text{actual}) = \frac{L}{P} \tag{12}$$

The results are shown in Fig. 5 which plots P versus L for refrigerative loads at temperatures of 77°K, 85°K and 90°K and which shows also the study limits for the refrigerative loads.

All the above data are conservative ratings of the power requirements based on current practice. It is anticipated that for the period 1965-1970 some improvement can be made, notably in motor and compressor efficiencies and a 10% improvement could be anticipated. It is considered that optimistic figures for the power requirements therefore would be smaller than those given above by about 10%.

2.13 <u>LIQUEFIERS</u>.

The simplest system for liquefaction consists of a compressor, cooler, heat exchanger, and expansion valve, provided the working gas in the cycle is the same as that to be liquefied.

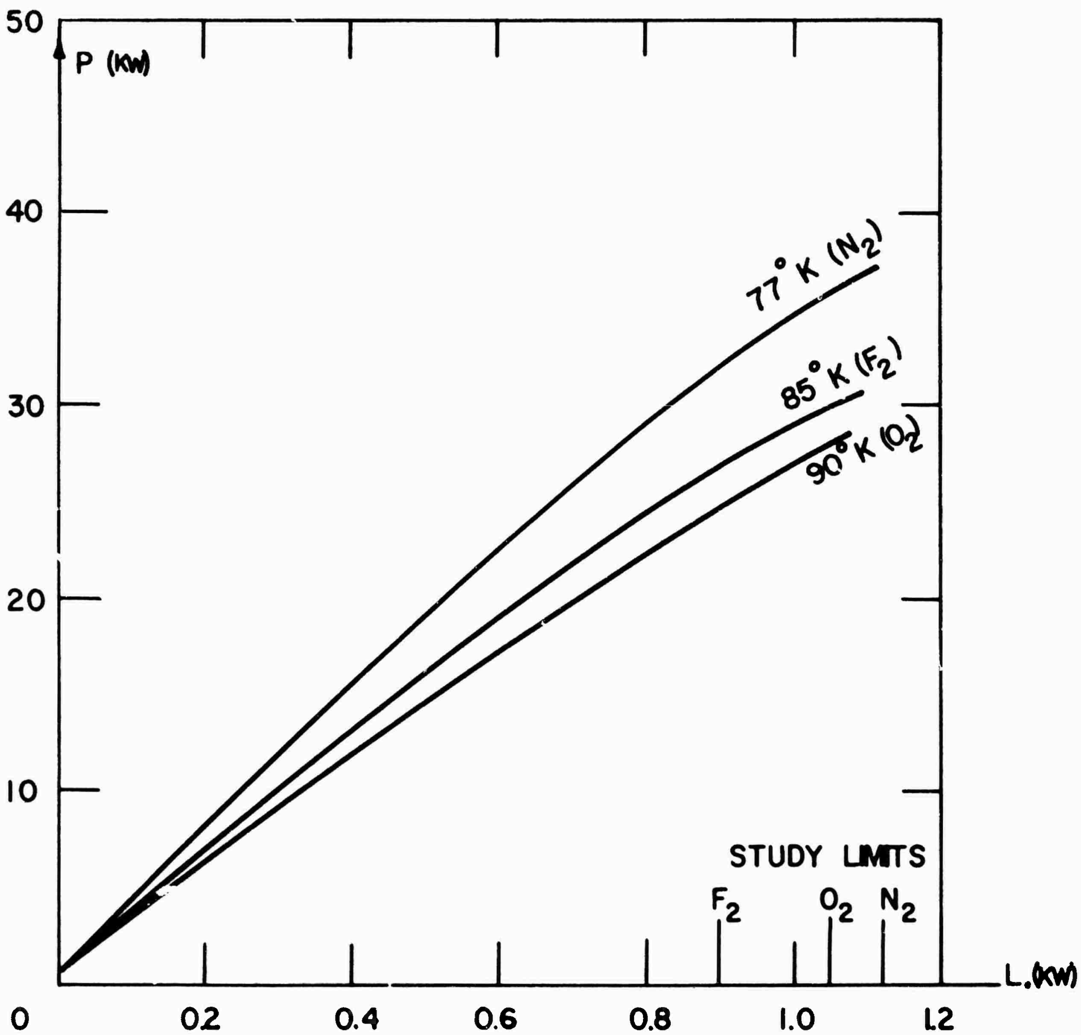

Fig. 5 - Power requirements (kW) versus refrigerative loads for single-stage J-T cycle refrigerators using N_2 as working fluid, (reliquefiers).

A system of this type is shown schematically in Fig. 6. After compression to pressure p_1 and subsequent cooling to the sink temperature, T_s, in the after-cooler, the fluid passes through the heat exchanger (path a-b) where it is cooled by the gas exhausted from the reservoir. The fluid is then expanded isenthalpically through the J-T valve (path b-c) and liquid is produced. This product liquid is removed through drain valve, and the cool gas at pressure p_2 and reservoir temperature, T, is exhausted through the heat exchanger (path d-e). This gas, to which makeup gas is added, is compressed, thus completing the cycle.

To calculate the liquefaction coefficient, ϵ, (the fraction liquefied) it is convenient first to assume ideally reversible conditions of operation, in which heat exchanger inefficiencies, heat losses, pressure drops in fluid flow, etc., are neglected. In this case we assume:

$$T_s = T_a = T_e; \; T = T_c = T_d; \quad p_1 = p_a = p_b; \; p_2 = p_c = p_d = p_e. \tag{13}$$

The fraction liquefied, ϵ, for such a system, in which the gas liquefied is the same as the working gas, is obtained from the enthalpy balance equation:

$$h_a = (1 - \epsilon)h_e + \epsilon h_f \tag{14}$$

or

$$\epsilon = \frac{h_e - h_a}{h_e - h_f} \tag{15}$$

where h_f is the enthalpy in joules/g of the liquid at p_2 and T. It can be shown* that for nitrogen liquefaction the work of compression represents very closely the work done in liquefaction. Again, if it is assumed that the compression is isothermal at the sink temperature, T_s, then the isothermal power of compression for a mass flow rate through the compressor of n g/sec is:

$$P_{iso} = nT_s\,(s_e - s_a) \approx (nRT_s/M)\ \ln\,(p_1/p_2) \text{ watts,} \tag{16}$$

where s is the entropy of the working fluid per g, M the molecular weight and R the gas constant in joules/g. This work results in liquefaction of $n\epsilon$ g/sec. The ideal isothermal power requirement per g liquefied therefore is:

$$P\text{ (ideal)} = \left(\frac{h_e - h_f}{h_e - h_a}\right) \cdot \frac{RT_s}{M}\ \ln\left(\frac{p_1}{p_2}\right) \text{ watts/(g. liquefied/sec).}$$

*Daunt, J. G., Handbuch der Physik 14, 1 (1956).

$$= 2.90 \times 10^{-4} \left(\frac{h_e - h_f}{h_e - h_a}\right) \cdot \frac{RT_s}{M} \cdot \log_{10}\left(\frac{p_1}{p_2}\right) \quad \text{kW-hr/lb.} \tag{17}$$

where R is the gas constant in joules/mole-°K.

To allow for non-isothermal compression, for motor and motor drive inefficiencies and for the non-ideality of the cycle, the above expression must be divided by $\xi\eta\zeta$ where ξ is the isothermal compressor efficiency, η is the relative efficiency of the actual cycle and ζ the motor efficiency.

For liquefaction of O_2 and F_2 one would not use exactly the system above described, since it would be hazardous and impractical to operate oxygen or fluorine compressors up to the high pressures (about 2400 psia) which are required for this cycle. In practice, as discussed above for single-stage J-T refrigerators, one would use N_2 as the working fluid, circulating in a closed loop in the system and use the refrigerative effect to condense the O_2 and F_2 at approximately atmospheric pressure. In addition, one would allow heat exchange between the returning low pressure N_2 gas in the cycle with the ingoing warm O_2 or F_2 gas. This scheme is shown diagrammatically in Fig. 7.

To calculate the fraction liquefied, ϵ, again it will first be assumed, as before, that ideally reversible processes occur such that:

$$T_s = T_a = T_e;\ T = T_c = T_d;\ p_1 = p_a = p_b;\ p_2 = p_c = p_d = p_3$$
$$\text{and } p = p_g = p_k. \tag{18}$$

Furthermore, we suppose that for every g of N_2 gas circulated in the closed loop, an amount ϵ in grams of O_2 or F_2 is condensed.

Then the enthalpy balance equation gives:

$$h_a + \epsilon h_g = h_e + \epsilon h_k \tag{19}$$

or

$$\epsilon = \frac{(h_e - h_a)}{(h_g - h_k)} \tag{20}$$

where h_e and h_a are the enthalpies in joules per g. of the nitrogen at e and a and where h_g and h_k are the enthalpies in joules/g of the O_2 or F_2 at g (T_s, p) and k (liquid at T and p).

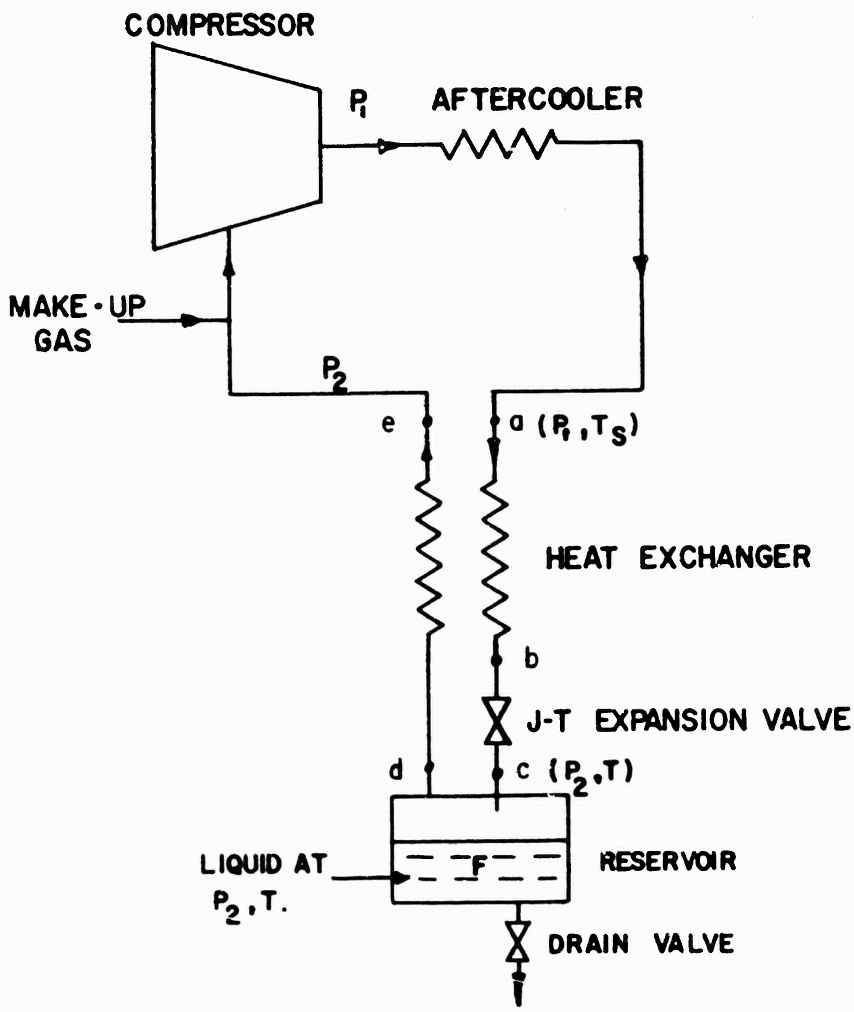

Fig. 6 - Flow diagram of single-stage J-T liquefier in which working gas is identical with gas liquefied.

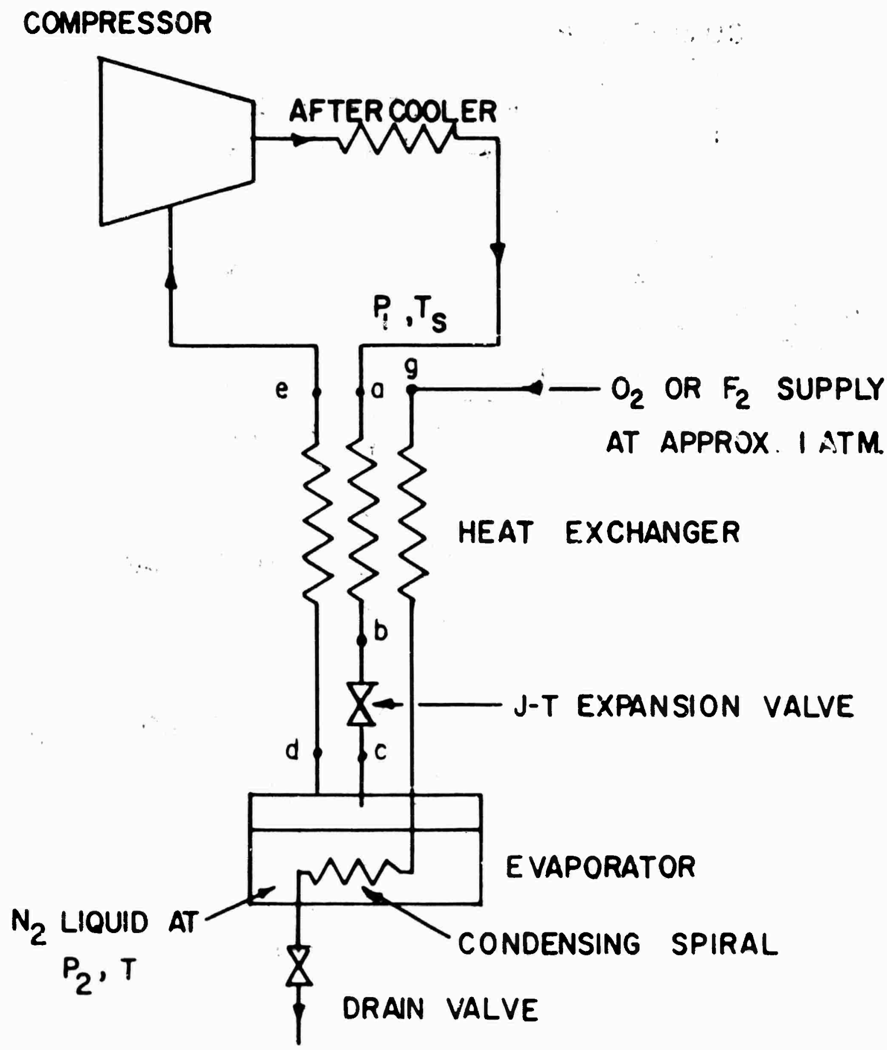

Fig. 7 - Flow diagram of a single-stage J-T liquefier in which working gas is different from gas liquefied.

Again, assuming isothermal compression and compressor conditions as for a perfect gas, we get for the isothermal power of compression for a nitrogen fluid flow of n g/sec:

$$p_{iso} \approx (nRT_s/M) \ln (p_1/p_2) \text{ watts} \tag{21}$$

where M is the molecular weight of nitrogen. The ideal power required for a rate of liquefaction of one g/s is:

$$P\ (\text{ideal}) \approx \frac{RT_s}{\varepsilon M} \ln\left(\frac{p_1}{p_2}\right) \text{ joules/g}$$

$$= \left(\frac{h_g - h_k}{h_e - h_a}\right) \cdot \frac{RT_s}{M} \ln\left(\frac{p_1}{p_2}\right) \text{ watts/(g. liquefied/sec)}$$

$$= 2.90 \times 10^{-4} \left(\frac{h_g - h_k}{h_e - h_a}\right) \frac{RT_s}{M_{N_2}} \log_{10}\left(\frac{p_1}{p_2}\right) \text{ kW-hr/lb.} \tag{22}$$

2.14 POWER REQUIREMENTS FOR SINGLE-STAGE J-T SYSTEMS FOR LIQUEFACTION OF N_2, O_2 and F_2.

As discussed in section V.2.12, for N_2 as the working fluid in the single-stage J-T liquefier, the compressor output should be taken as 2400 psia. As before, T_s will be taken to be 300°K. The pressure p_2 will be a function of the temperature T in the evaporator, being determined by the vapor pressure of N_2. The data required to calculate P (ideal) and the results are shown in Table 9.

P (actual) is obtained by putting ξ, the isothermal compressor efficiency for a diatomic gas (N = 1.4) = 67%, the motor efficiency, ς , for higher level operation (~ 1000 lbs/day) at 85% and at 80% for low level operation (~100 lbs/day) and by putting the relative efficiency of the actual cycle, η, (see Section V.2.12) at 85% and 80% for the higher level and low level operations respectively. The data for C.P. (actual) under these conditions are presented in Table 10.

TABLE 9

DATA FOR CALCULATION OF POWER REQUIREMENTS FOR IDEAL SINGLE-STAGE J-T CYCLE LIQUEFIERS FOR N_2, O_2 AND F_2

Gas	B.P. °K	p_2 atmos.	$(h_e-h_a)N_2$ joules/g	$(h_g-h_k)O_2$ or F_2 or $(h_e-h_f)N_2$ joules/g	ε %	P(ideal) kW-hr/lb.
N_2	77	1	27.6	435	6.35	0.90
F_2	85	2.4	26.8	345	7.75	0.61
O_2	90	3.6	26.1	405	6.36	0.67

TABLE 10

ACTUAL POWER REQUIREMENTS FOR SINGLE-STAGE J-T CYCLE LIQUEFIERS OF N_2, O_2 AND F_2

Gas	B.P.	P (ideal)	P (actual)	
			Low Level	Higher Level
	°K	kW-hr/lb.	(100 lbs/day) kW-hr/lb.	(1000 lbs/day) kW-hr/lb.
N_2	77	0.90	2.1	1.87
F_2	85	0.61	1.42	1.26
O_2	90	0.67	1.56	1.38

From the above data the curves of Fig. 8 were drawn, which show the power P required for liquefaction as a function of the liquefaction rate in liters/hr for N_2, F_2 and O_2. Also shown on the graph are the study limits.

All the above data are conservative ratings of the power requirements based on current practice. As mentioned in Section V.2.12 above, optimistic ratings for the period 1965-70 would be about 10% less than the above figures.

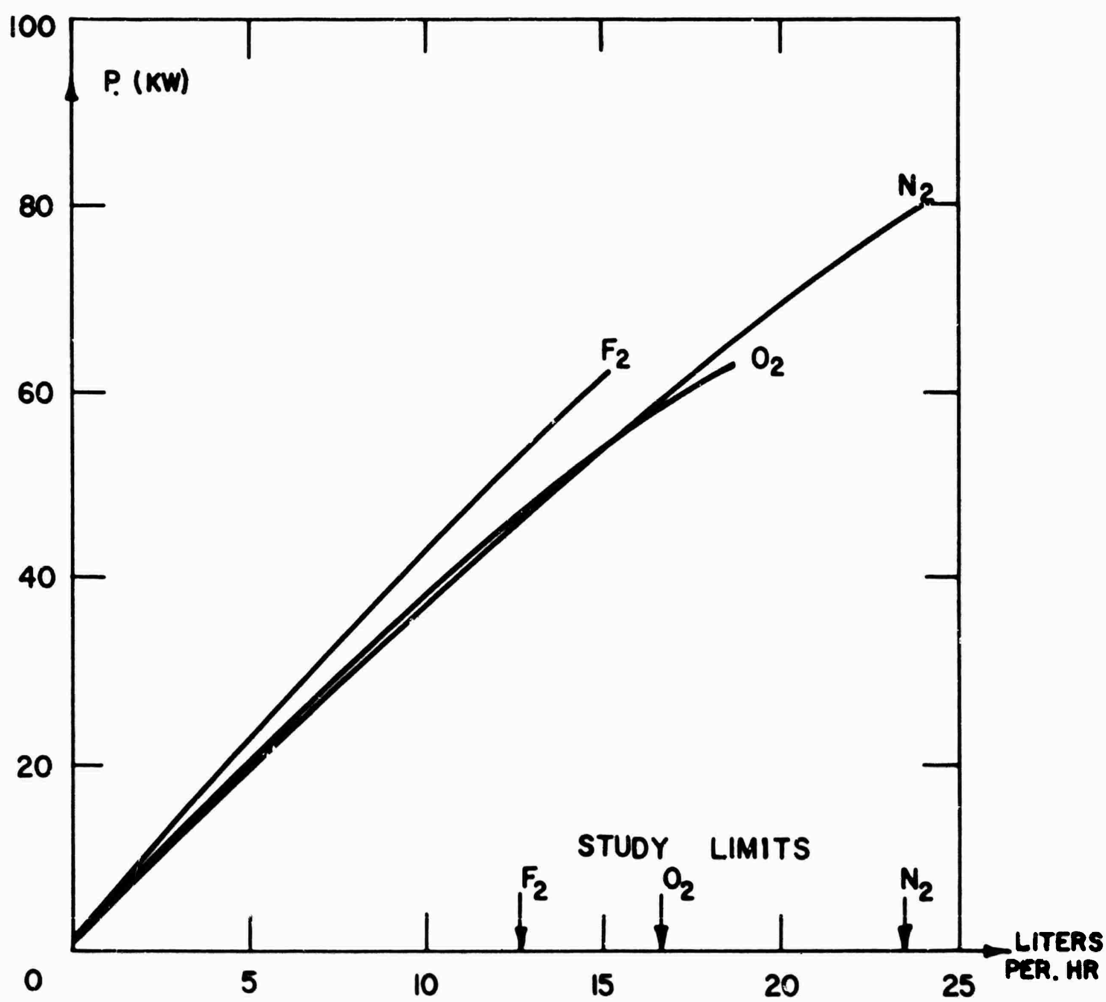

Fig. 8 - Power requirements (kW) for liquefaction of N_2, F_2, and O_2, using single-stage J-T cycle systems (see text) versus rate of liquefaction, (liters/hr.).

2.2 PRECOOLED SINGLE-STAGE JOULE-THOMSON SYSTEMS FOR OPERATION AT TEMPERATURES DOWN TO 77°K

2.21 REFRIGERATION (RELIQUEFACTION) DOWN TO 77°K USING PRE-COOLED SINGLE-STAGE J-T SYSTEMS.

A schematic diagram of a precooled single-stage J-T refrigerating system is shown in Fig. 9. The fluid enters the first heat exchanger at the point, m, after compression to pressure p_1 and after cooling in the after-cooler (radiator) to the sink temperature, T_S. It passes through the refrigerator-evaporator which is maintained at an intermediate temperature, T_i, by a secondary refrigerator (generally a two-stage freon compressed vapor refrigerator) and the fluid reaches the point "a" at the temperature, T_i. Then the fluid passes through the J-T heat exchanger (path a-b). where it is cooled by the gas from the final evaporator. The path a-b-c-d-e is identical with the process illustrated in Fig. 3. The return fluid at e passes back through the first heat exchanger to r, where it enters the compressor for recompression. The liquid evaporating in the final evaporator serves to counterbalance heat load, L, at temperature T.

To discuss the operation of this system, it will first be assumed that the cycle has no irreversibilities, i.e., it will be assumed that:

$$T_s = T_m = T_r; \quad T_i = T_a = T_e; \quad T = T_c = T_d \tag{23}$$

$$p_1 = p_m = p_n = p_a = p_b; \quad p_2 = p_c = p_d = p_e = p_r \tag{24}$$

Then, as shown in Section V.2.11, the ideal coefficient of performance is:

$$\text{C.P. (ideal)} = \frac{L}{p_{iso}} \approx \frac{(h_e - h_a)}{(RT_s/M)\ \ln\ (p_1/p_2) + p_i} \tag{25}$$

where the symbols have their previously designated significance and where p_i is the power required to operate the secondary refrigerator at T_i. Since the actual coefficient of performance of such secondary refrigerators can be quite high, approaching the Carnot values for cascaded compressed vapor refrigerators, there is a net gain in the process over the J-T system without precooling. To understand this, some typical values of $(h_e - h_a)$ for N_2 are listed in Table 11 below for various values of the intermediate temperature T_i. These data are for $p_1 = 2400$ psia and $p_2 = 1$ atmos.

TABLE 11

SOME ENTHALPY VALUES FOR N_2

T °K	$(h_e - h_a)$ joules/g
300	27.6
290	29.9
280	32.2
270	35.2
260	36.7
250	40.4
240	44.1
230	47.9

If one neglected p_1 therefore, the C.P. (ideal) of the system with N_2 as the working gas would be enhanced by a factor (47.9/27.6) = 1.73 by reducing T_1 from 300°K (as for a system without precooling) to 230°K, which latter temperature can readily be maintained by cascaded compressed vapor refrigerators. It is beyond the scope of this report to calculate in detail the actual coefficients of performance which can be obtained with precooled single-stage systems, since this would require a detailed knowledge of the heat exchanger efficiencies for both interchangers, the compressed-vapor refrigerator efficiencies, etc. An estimate of the improvement can be made by a comparison of the known performance of current equivalent liquefiers, using single-stage J-T cycles with and without precooling in the same refrigerative load range as that covered in this report. These data* indicate that by precooling to approximately 250°K the overall actual coefficient of performance, C.P. (actual), can be increased by a factor of approximately 1.5 over the values given in Section V.2.12 for

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*Data taken from: Greenwood, H. C., Industrial Gases, Van Nostrand (1919); Linde, R. Z., VDI 65, 1357 (1921); Lenz, H., Hdb. der Exp. Phys. 9/1, 127 (1929); Davies, M., Gas Liquefaction and Rectification, (Oxford Press, 1940); Hansen, H., Hdb. d. Kaltetech. 8, 1 (1957); Daunt, J. G., Hdb. der Physik 14, 1 (1956).

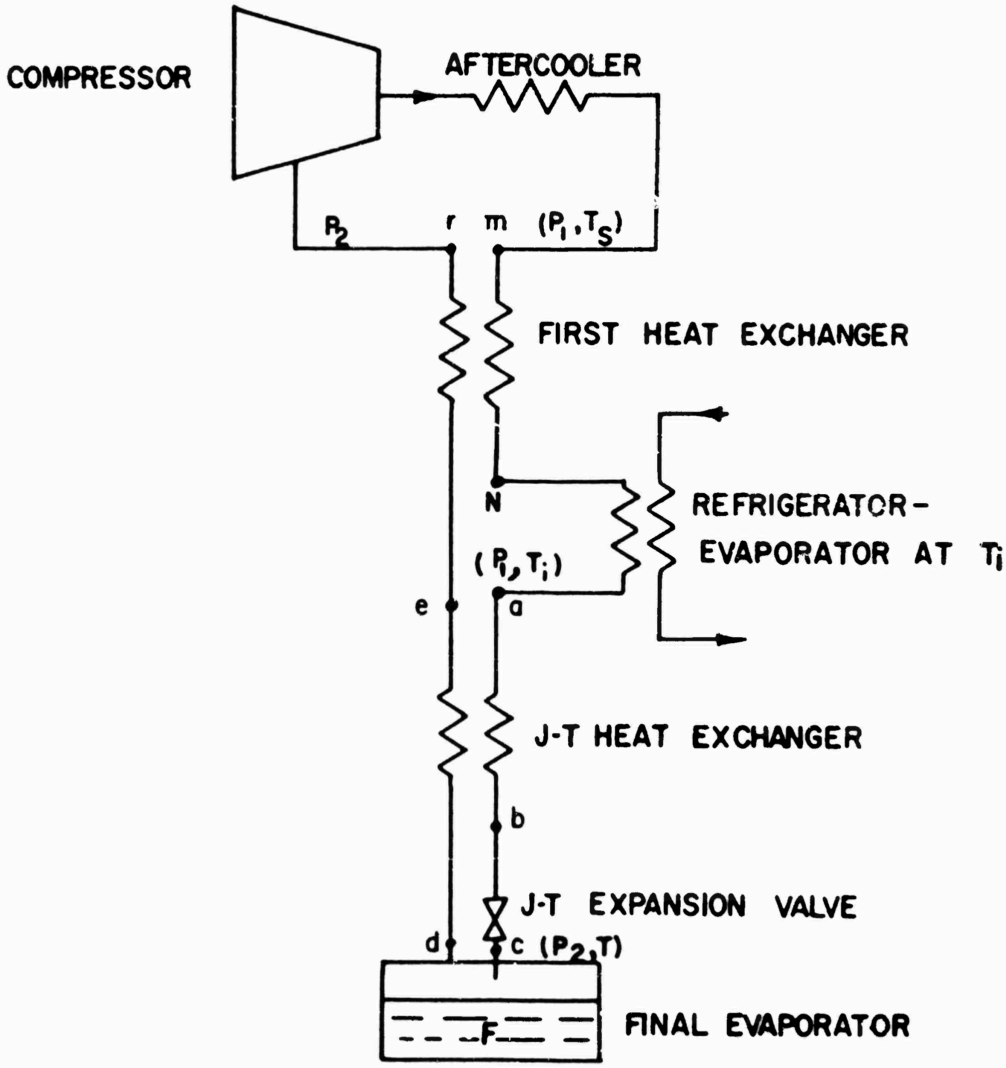

Fig. 9 - Schematic diagram of a precooled single-stage J-T refrigerating system.

the single-stage J-T cycle without precooling. In the final comparison of cycles, therefore (see Section VIII), this factor will be taken into account. It must be noted, however, that by the introduction of precooling refrigeration, the overall system is made more complex, leading to reduced reliability. Since, however, vapor compression machines, for the precooling temperatures considered, are highly reliable in operation, this factor may not be too serious, provided they are suitable for space operation. Moreover, since the major weight and volume of the overall system will reside in the N_2 compressor for the main working cycle, a major reduction in the magnitude of these parameters would result from precooling.

2.22 LIQUEFACTION OF N_2, F_2 AND O_2 USING PRE-COOLED SINGLE-STAGE J-T CYCLES.

The flow diagram for an N_2 liquefier using a precooled single-stage J-T cycle would be identical with that for the equivalent refrigerator shown in Fig. 9 except for the provision of a drain-valve to tap off the liquid from the reservoir (the evaporator of Fig. 9) and for the provision of make-up gas to the input of the compressor. A pre-cooled single-stage J-T cycle for O_2 or N_2 liquefaction would have a flow diagram similar to that for the single-stage J-T cycle without precooling given in Fig. 7 for the flow pattern below the intermediate temperature. Above the intermediate temperature the flow diagram would include the condensor-evaporator at temperature T_1 and a first interchanger which would exchange heat between the ingoing high-pressure nitrogen working gas, the ingoing low pressure O_2 or F_2 gas operating in a separate condensing circuit, and the outgoing cold return nitrogen working gas.

To gain insight into the increased yields available by precooling, the overall reduction in the power requirement, P (actual), for an actual liquefier liquefying, say, N_2, F_2 or O_2 in the liquefaction range of interest to this study would be by a factor of approximately $1/1.5 = 0.67$, if precooling is carried out at a readily approachable temperature of 250°K.* This factor will be used in the comparison of cycles in Section VIII. The comments made above in Section V.2.21 with regard to weight and volume of precooled refrigerating systems apply also to precooled liquefiers for O_2, F_2 and N_2.

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*Daunt, J.G. Hdb. d. Physik. 14. 1 (1956)

2.3 TWO-STAGE JOULE-THOMSON PROCESS FOR LIQUEFACTION AND RE-LIQUEFACTION DOWN to 77°K

There is no reference in the literature to a closed-cycle refrigerator utilizing a two-stage Joule-Thomson process, and in consequence it is advantageous to consider first the known liquefaction systems using this cycle and to use the results from the data on such liquefiers for assessment of the performance of refrigerators.

Fig. 10 shows a flow sheet of a two-stage J-T cycle for liquefaction of gases such as air and nitrogen. The compression is made up in two main stages so that the intermediate pressure return gas, at p_2, can feed into the compressor between stages. After passing through the three circuit heat interchanger, the high pressure gas at pressure p_3 expands in the first Joule-Thomson valve, V_1, to an intermediate pressure p_2. A fraction x of this expanded gas is further expanded through the valve, V_2, into the reservoir at low pressure (p_1) and temperature T. The remaining fraction (1-x) returns via the three circuit heat interchanger to the compressor at the intermediate pressure p_2. That fluid which expands at V_2 partially liquefies. The liquid collects in the reservoir at temperature T and can be drained off through the drain valve. For a refrigerator the drain valve would be omitted and the lower reservoir would then act as an evaporator, liquid evaporating therefrom providing the refrigeration. It would be possible theoretically for such a refrigerating system to provide refrigeration at two temperature levels, that is, both in the intermediate reservoir at T_1 and in the evaporator. The system described would be suitable for reliquefaction of oxygen, fluorine and nitrogen using preferably nitrogen as the working fluid refrigerant.

Fig. 11 shows the cycle on a T-S diagram when employed for liquefaction of the same gas as the working fluid. In this diagram the lettered points correspond with those on the diagram of Fig. 10. Also in the T-S diagram, for simplicity the compression (path a-b-c) has been assumed to be isothermal.

According to Lenz*, in practice the two-stage J-T cycle for air liquefaction without precooling operating with p_3 = 200 atm., p_2 = 50 atm. and x = 0.2, yields a practical minimum coefficient of performance of about 1.15 kW-hr/kg liquefied, which is approximately twice as efficient as a single-stage J-T cycle without precooling operating at the same high pressure, p_3 = 200 atm. (x is the fraction of fluid passing to the second J-T valve).

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*Lenz, H., Hdb. der Exp. Phys. 9/1, 127 (1929).

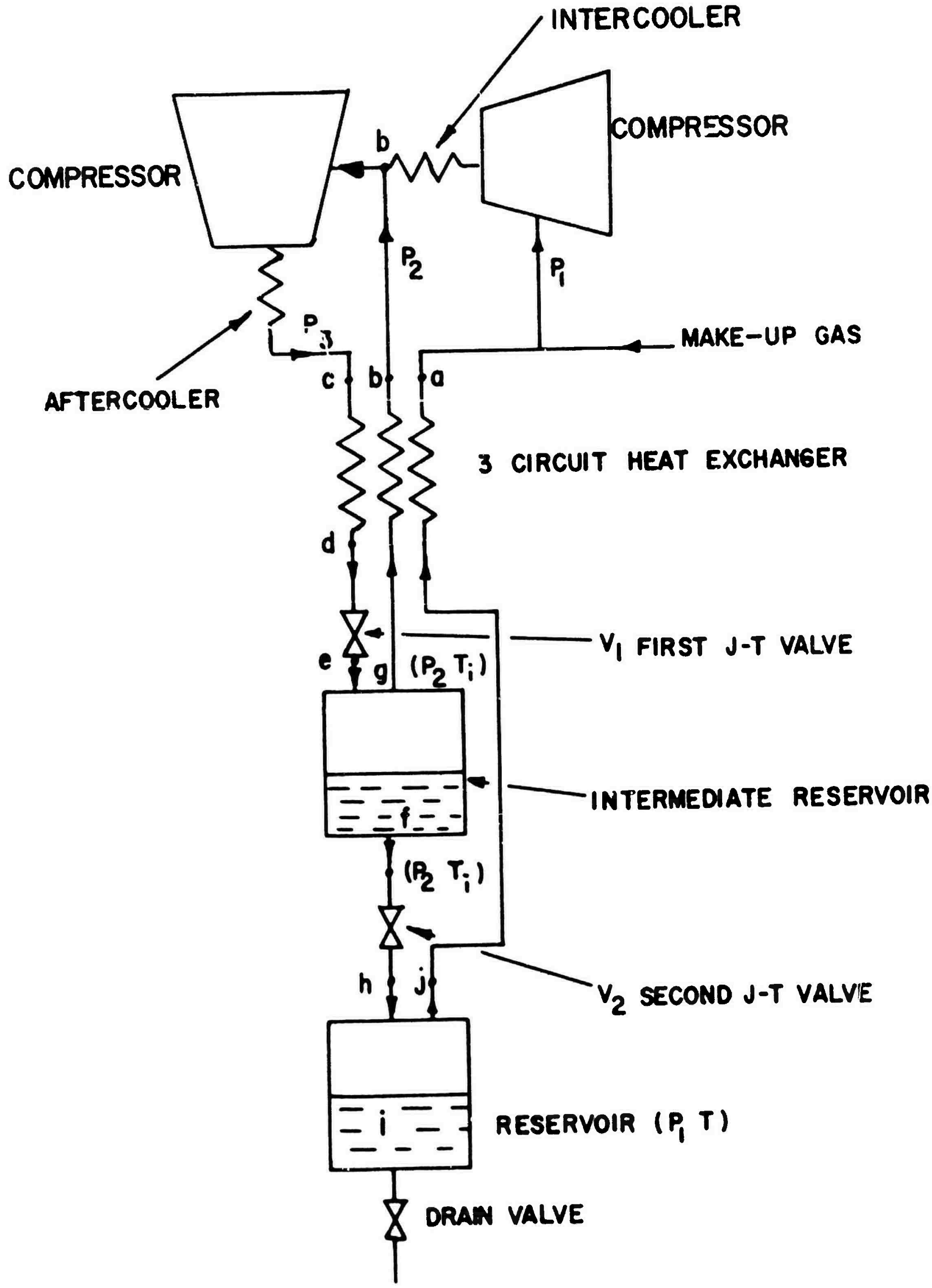

Fig. 10 - Schematic diagram of two-stage J-T cycle for liquefaction of gases such as air or nitrogen.

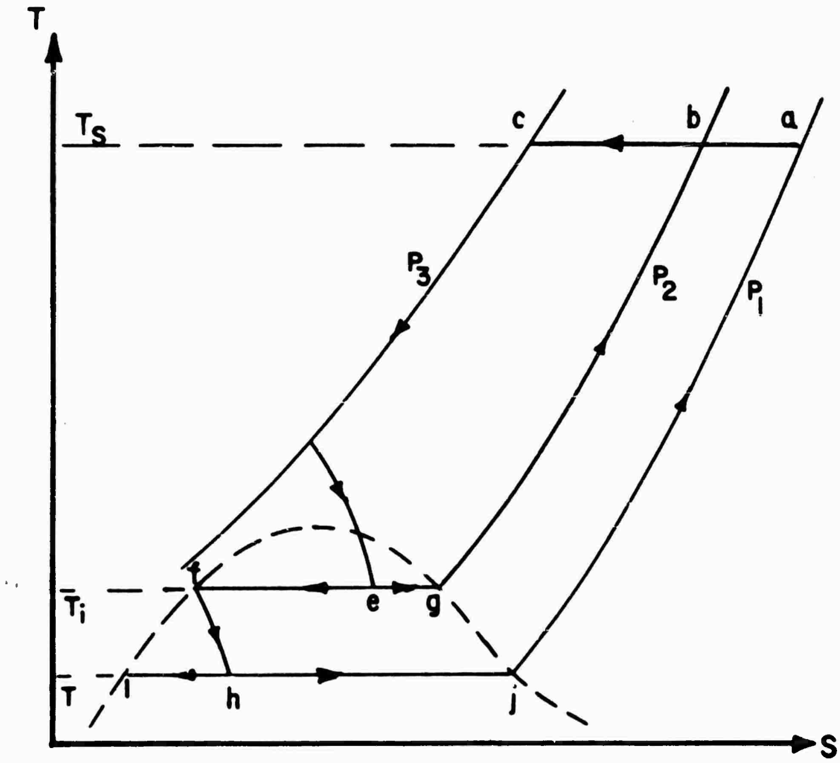

Fig. 11 - T-S diagram of liquefaction by two-stage Joule-Thomson Process.

For a two-stage air liquefying precooled system, according to Lenz,* the power requirement for $x = 0.4$, $p_3 = 200$ atm. and $p_2 = 40$ atmos. and a precooling temperature of 223°K is approximately 0.65 kW-hr/kg of air liquefied, and under similar conditions at a precooling temperature of 250°K, the minimum power requirement is approximately 0.92 kW-hr/kg.

The reduction in the power requirement for this air liquefying system without precooling, obtained by going from a single-stage J-T cycle to an optimized double-stage J-T cycle, is approximately by a factor of 0.525 and the corresponding factor for the precooled systems with a precooling temperature of 250°K is approximately 0.62. Although the actual values of these reductions depend on the characteristics of the gas used as the working fluid, it has been estimated that they will be approximately the same for use of N_2 as the working fluid. In the comparison of cycles, given in Section VIII, therefore, these ratios will be used in conjunction with the data given in Sections V.2.12 and V.2.13 for estimating the actual coefficient of performance of liquefiers and reliquefiers based on double-stage precooled and non-precooled J-T cycles and operating in the range of interest to this report.

2.4 RELIQUEFACTION OF H_2 AND He USING PRECOOLED J-T SYSTEMS

A schematic arrangement shown in Fig. 12 is of a typical closed-cycle hydrogen refrigerator,** in which a liquid nitrogen bath is used for precooling. The compressed hydrogen is passed through the three way H_2-H_2-N_2 heat exchanger, where it is cooled against evaporated N_2 and exhaust low pressure H_2. It is then cooled further in liquid nitrogen and in the final heat exchanger against the hydrogen vapor from the evaporation. Liquid is then formed in the J-T expansion valve. The liquid is evaporated in the evaporator against the refrigerated load. The exhaust gas is used to cool the incoming high pressure hydrogen and is then recompressed, so completing the cycle.

Operating data for a system such as that of Fig. 12 built by the Cryogenic Engineering Laboratory, NBS***, are such that a total refrigeration of 329 watts at 27°K is available for a

*Lenz, H., Hdb. der Exp. Phys. 9/1, 127 (1929).

**See McIntosh, Mann, Macinko and van der Arend. Adv. Cryo. Eng. 1.62.1960. See also Birmingham, Chelton, Mann and Hernandez. ASTM Bulletin No. 240. p. 34. 1959; Chelton, D. B. and Dean, J. W. NBS Tech. Note No. 38.1960 and Chelton et al. NBS Tech. Note No. 39. Jan. 1960.

***See Chelton, Dean and Birmingham. Rev. Sci. Instr. 31. 712. 1960.

hydrogen flow of 35 scfm at an input pressure of 1800 psig and a precoolant temperature of 65°K. Under these conditions the liquid N_2 consumption was 8.3 liters/hr.

Figs. 13 and 14 (taken from Chelton et al*) give the H_2 compressor power requirements and the liquid nitrogen consumption for hydrogen refrigerators of the type shown in Fig. 12. For Fig. 13 the isothermal compressor efficiency was assumed to be 65%. To make the data presented in Fig. 13 comparable with that given elsewhere in this report, it should be modified to allow for 67% isothermal compressor efficiency and to include 90% combined motor and motor-drive efficiency. Making these modifications and assuming the H_2 high pressure input pressure to be 120 atm., one obtains from Fig. 13 for the power of compression per kW refrigeration the following: 40 KW for 21°K refrigeration and 57 kW for 15°K refrigeration. It is to be noted that these data were obtained on the assumption that the efficiency of the final J-T heat exchanger was 100%.

The lower limit of the liquid nitrogen consumption in the precooler is determined by the ΔT at the warm end of the three-way H_2-H_2-N_2 heat exchanger between the outgoing N_2 stream and the ingoing high pressure H_2. This ΔT is the ΔT_4 of Fig. 14. It is to be noted that the liquid N_2 consumption, as shown in Fig. 14, is independent of the hydrogen evaporator temperature. If one assumes that ΔT_4 is 10°K and ΔT_3 is zero (ΔT_3 is the temperature difference between the ingoing and outgoing H_2 streams at the warm end of the final J-T heat exchanger) then from Fig. 14 the liquid N_2 consumption is 20.6 liters/hr per kW of H_2 refrigeration. Taking a value of 0.69 kW-hr/lb for N_2 liquefaction for a relatively small scale system (see subsection 3.14), one obtains for the power of liquid N_2 liquefaction 25.4 kW per KW of refrigeration at hydrogen temperatures. The total minimum power for refrigeration at 21°K, using this system, is therefore 65.4 kW per kW load and 82.4 kW per kW load at 15°K. The corresponding maximum coefficients of performance, therefore, are: 0.015 at 21°K and 0.012 at 15°K and, by interpolation, 0.0145 at 20°K. However the power required for pumping the liquid N_2 to the precooling temperature of 65°K is not included here.

Fig. 15 shows a schematic diagram of the precooled J-T hydrogen refrigerator described above used as a precooling medium for bringing helium below its inversion temperature.** As shown, the compressed helium is cooled against evaporated nitrogen and exhaust low pressure helium. It is further cooled

*Chelton, D.B., et al. NBS Tech. Note No. 39. 1960.

**See Chelton, Dean, Strobridge, Birmingham and Mann. NBS Tech Note No. 39. 1960.

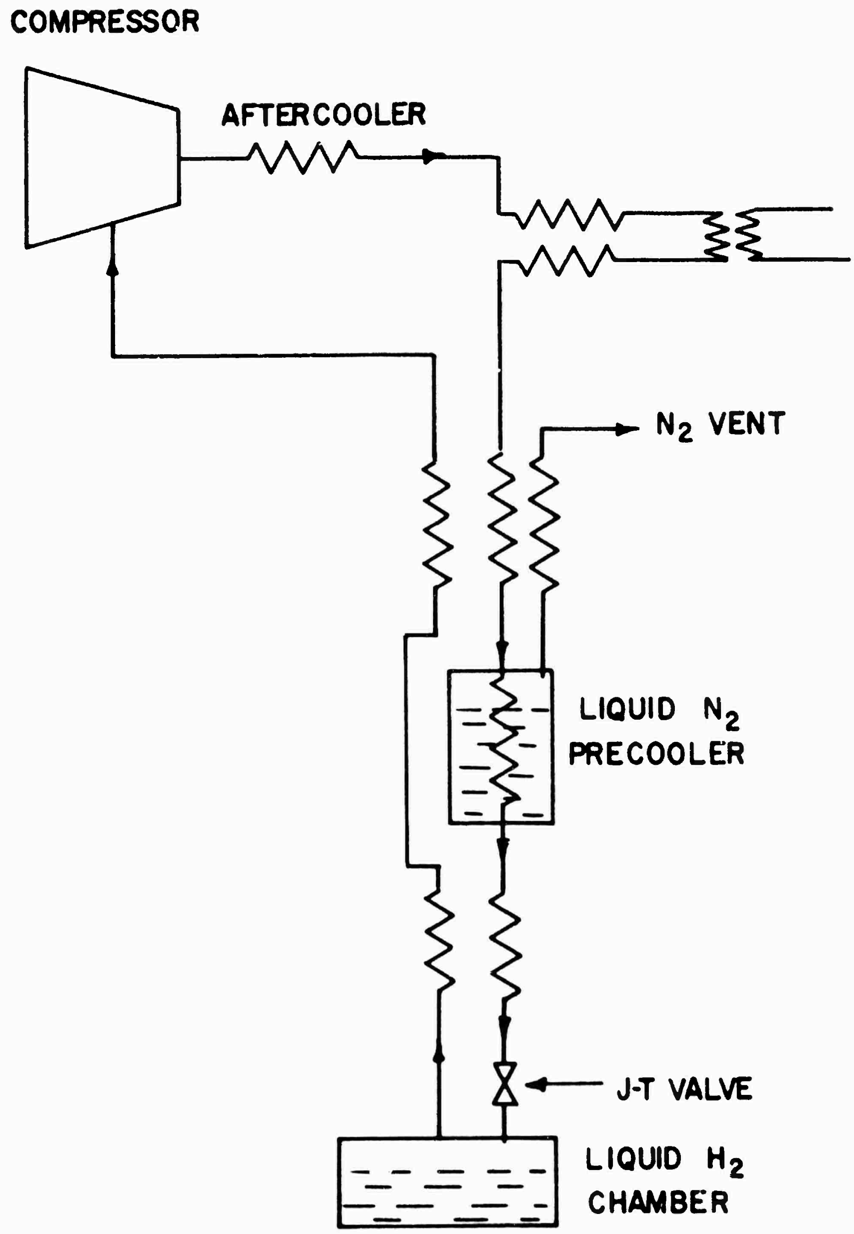

Fig. 12 - Flow diagram of precooled J-T hydrogen refrigerator.

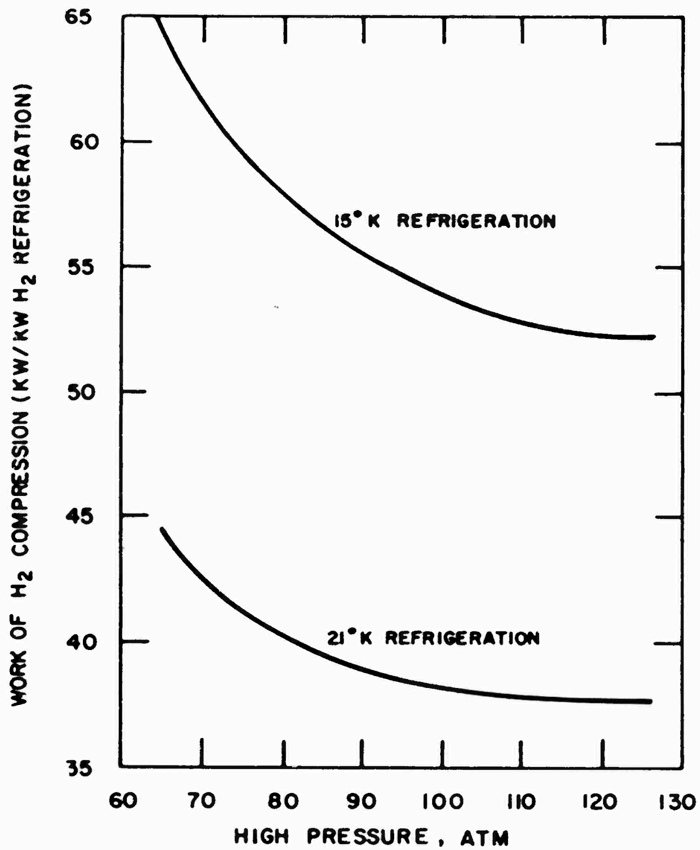

Fig. 13 - *Work of H_2 compression in precooled J-T H_2 refrigerator, assuming isothermal compressor efficiency to be 65%, according to D. B. Chelton, et al, (NBS Tech. Note No. 39, Jan. 1960).*

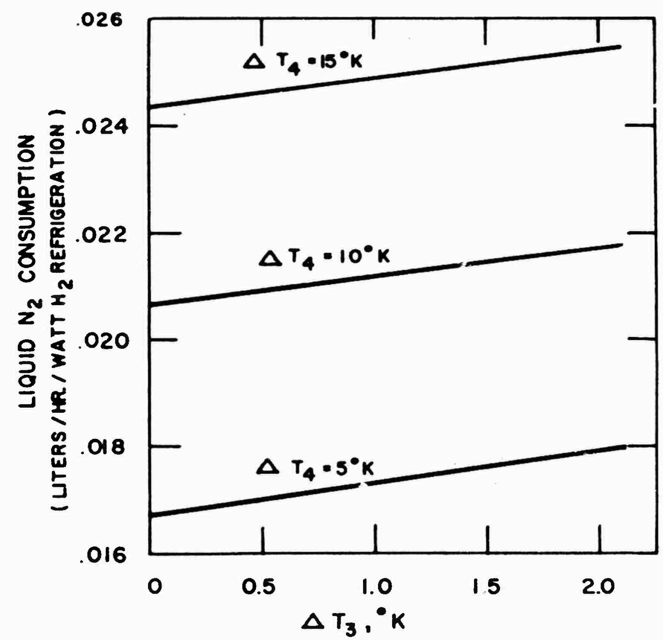

Fig. 14 - *Liquid N_2 consumption of precooled J-T H_2 refrigerator, according to D. B. Chelton, et al, NBS Tech. Note No. 39, Jan. 1960. (For definitions of T_3 and T_4, see text.)*

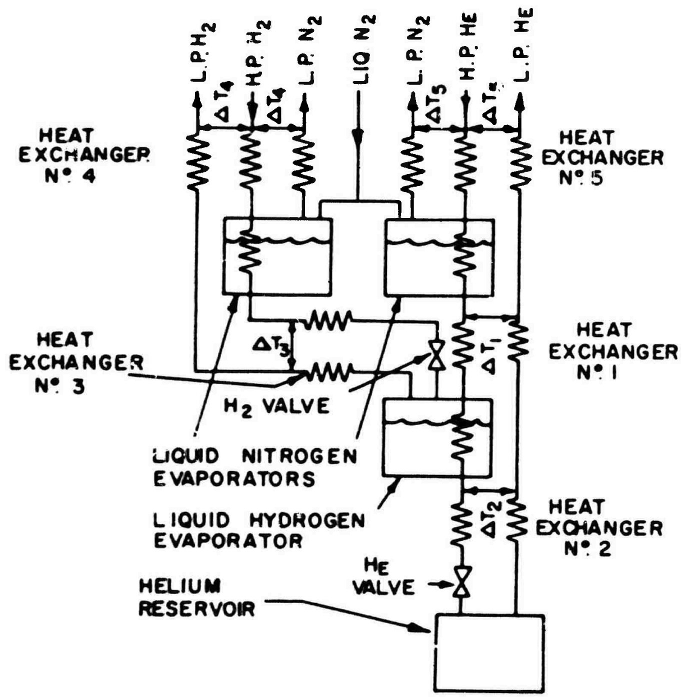

Fig. 15 - Schematic flow diagram of precooled J-T helium refrigerator, according to D. B. Chelton, et al, (NBS Tech. Note No. 39, 1960).

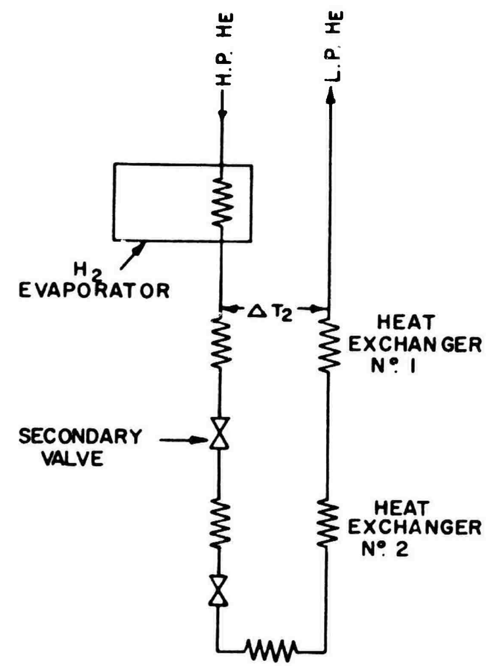

Fig. 16 - Modified flow diagram of J-T helium refrigerator, according to D. B. Chelton, et al, (NBS Tech. Note No. 39, 1960). The expansion is carried out in two stages in order to compensate, in the cold end heat exchanger, for the variation in specific heat of the high pressure gas.

against liquid nitrogen and then against low pressure helium in heat exchanger No. 1. After being cooled with boiling liquid hydrogen, the high pressure stream is cooled with boil-off from the helium evaporator, before being expanded in the throttle valve. Liquid is formed in the throttle valve and supplies refrigeration at liquid helium temperature by evaporating. The exhaust gas is used to cool high pressure helium and is then compressed, so completing the cycle. It was found, upon drawing the cooling curve for heat exchanger No. 2, that it was desirable to carry out the helium expansion in two stages, thus modifying the schematic diagram as shown in Fig. 16. The helium compressor power requirements for this modified helium refrigerator as a function of the inlet pressure are shown in Fig. 17 for two different H_2 precooling temperatures. In the evaluation of these power requirements, Chelton et al.* assumed 100% efficiency for the final J-T helium heat exchanger (heat exchanger No. 2 of Fig. 15). From Fig. 17 it will be seen that the minimum work of compression occurs for a helium inlet pressure of about 30 atm. This pressure, with a H_2 precooling temperature of 21°K, will be assumed in the subsequent calculations. These assumptions result in a power for compression of 384 kW per kW of refrigeration at helium temperature (normalized to 67% isothermal compressor efficiency and 90% motor and motor-drive efficiency).

Fig. 18 gives the results obtained by Chelton et al.* for the liquid N_2 consumption in the helium circuit of the J-T refrigerator as a function of the temperature difference, ΔT_2, at the warm end of heat exchanger No. 1 for various values of ΔT_5 (see Fig. 15 for the location of these ΔT's in the flow diagram). The liquid N_2 consumption is expressed in liter/hr per helium flow rate expressed in scfm. Fig. 19 gives this evaluation of the required helium flow rate in scfm per watt of refrigeration at helium temperatures as a function of ΔT_2 for two different temperatures of H_2 precooling. Assuming ΔT_2 to be zero and a precooling temperature of 21°K, the flow rate is 1.3 scfm per watt helium temperature refrigeration and this requires a liquid nitrogen consumption in the helium circuit of 73 liters/hr per kW helium temperature refrigeration, if ΔT_1 is zero and ΔT_5 is 10°K. Using, as before, a value of 0.69 kW-hr/lb for N_2 liquefaction we get the N_2 liquefaction power requirement for the helium circuit to be 89.5 kW per kW refrigeration at helium temperature.

Fig. 20 also taken from the same work by Chelton et al.,* gives the ratio of the helium temperature refrigerative load, Q_2, to the hydrogen temperature (21°K) refrigeration requirements Q_1

*See Chelton, Dean, Strobridge, Birmingham and Mann. NBS Tech Note No. 39. 1960.

as a function of ΔT_2 and ΔT_1. Assuming as before that ΔT_2 is zero and ΔT_1 is 1°K, this graph indicates that Q_2/Q_1 is 0.525; or that per kW refrigeration at 4.2°K, the hydrogen evaporator must supply 1.98 kW of refrigeration at 21°K. Using the data deduced above for the precooled hydrogen J-T refrigerator, one obtains a power requirement in the H_2 circuit of 132 kW minimum per kW of refrigeration at 4.2°K. Adding all the contributions to the total power, therefore, for the helium refrigerator, due to the helium compressor power, N_2 consumption in the helium circuit and to the H_2 circuit power requirements, one obtains an overall maximum coefficient of performance of 0.00165 for a precooling temperature of 21°K. As indicated in Fig. 17, considerable improvement is to be expected by going to lower precooling temperatures.

2.5 PRECOOLED J-T SYSTEMS FOR H_2 AND He LIQUEFACTION

Flow sheets for hydrogen and helium liquefaction with liquid nitrogen precooling are identical to those for hydrogen and helium refrigerators as shown in Figs. 12 and 15. The only difference is that the liquid formed in the J-T expansion valve is drawn off as a product rather than evaporated to provide refrigeration.

Figs. 21 and 22, taken from the work of Chelton et al. previously cited, show the work of H_2 compression as a function of the inlet pressure and precoolant temperature and the total work of H_2 liquefaction including precoolant. As is noted in section 3.42 below, in evaluating the total work of liquefaction it was assumed by Chelton et al. that the N_2 liquefaction used 0.284 kW-hr/lb. This figure is applicable to very large systems, whereas for systems of the size considered in this report it is considered that one should assume 0.69 kW-hr/lb for liquefaction of N_2 (see section 3.14). In order to allow for this, the following calculations have been made: from Fig. 21 the work of H_2 compression at an inlet pressure of 120 atm. is 2.67 kW-hr/gal for normal H_2 liquefaction and 3.45 kW-hr/gal for liquefaction and conversion to 95% para-H_2 when ΔT is 1°K and the precooling temperature 65°K. Normalizing these values to 67% isothermal compressor efficiency and 90% combined motor and motor-drive efficiency, one gets respectively 4.9 kW-hr/lb and 6.3 kW-hr/lb and the power for liquid N_2 production is respectively (assuming 0.69 kW-hr/lb for N_2 liquefaction) 4.2 kW-hr and 4.75 kW-hr per lb of H_2 liquefied. Added together this gives 9.1 kW-hr/lb of normal H_2 liquefied and 11.05 kW-hr/lb for liquefaction and conversion to 95% para-H_2.

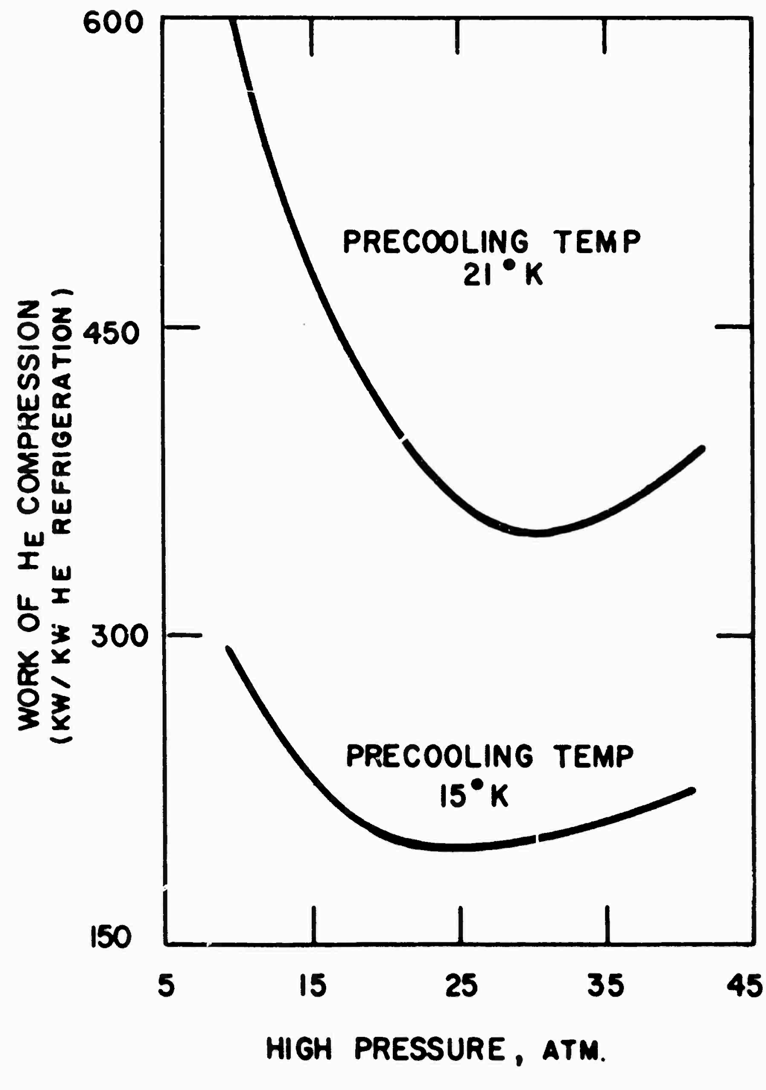

Fig. 17 - Work of helium compression in precooled J-T helium refrigerator, according to D. B. Chelton, et al, (NBS Tech. Note No. 39, 1960).

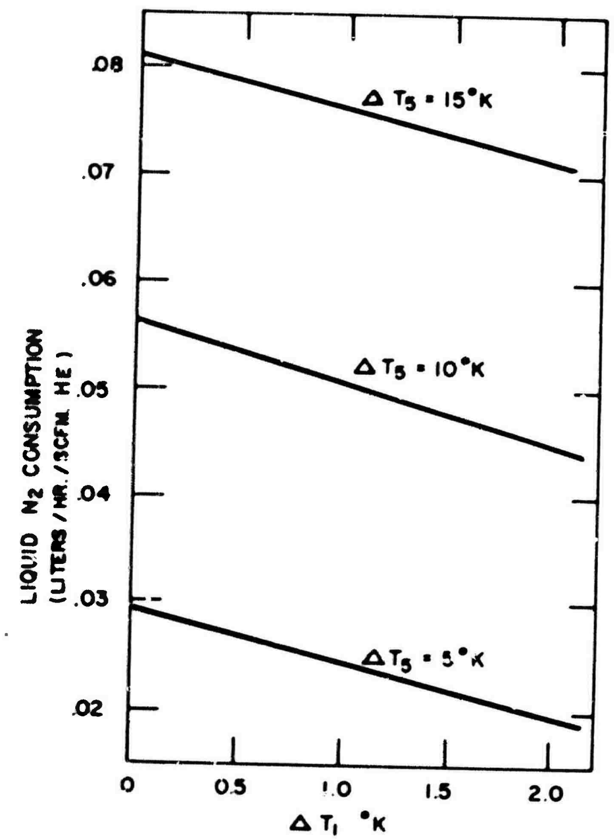

Fig. 18 - Liquid nitrogen consumption in the helium circuit of a precooled J-T helium refrigerator, according to D. B. Chelton, et al, (NBS Tech. Note No. 39, 1960).

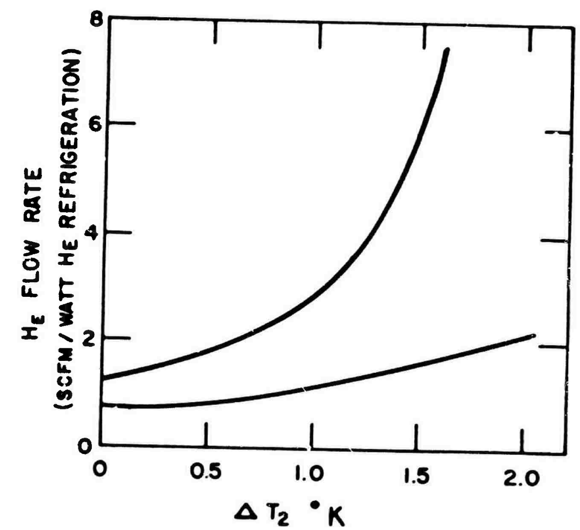

Fig. 19 - Helium flow rate per watt refrigeration in precooled J-T helium refrigerator, according to D. B. Chelton, et al, (NBS Tech. Note No. 39, 1960).

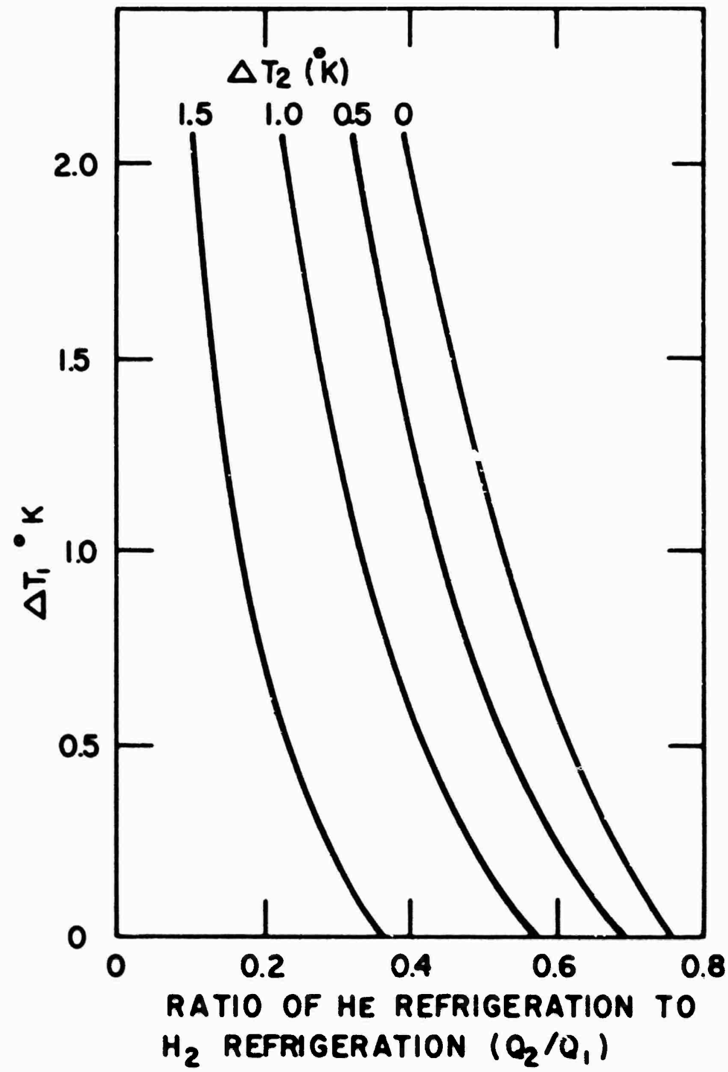

Fig. 20 - Helium refrigerator-effect of heat exchanger performance; 21°K precooling, according to D. B. Chelton, et al, (NBS Tech. Note No. 39, 1960).

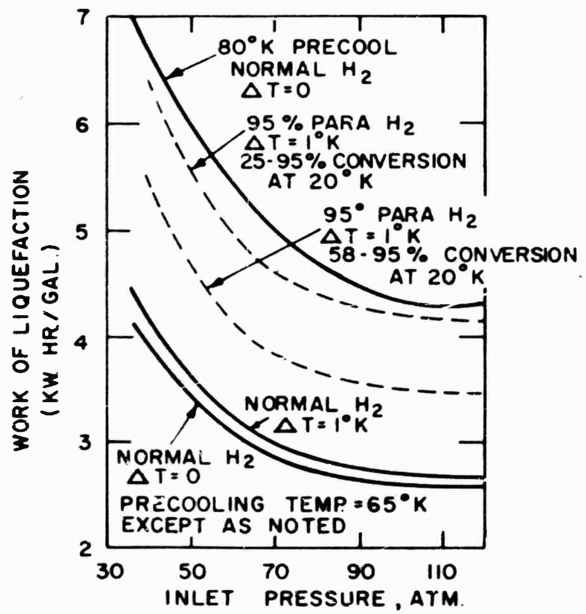

Fig. 21 - Work of H_2 compression of precooled J-T H_2 liquefier as a function of H_2 pressure and precoolant temperature, according to D. B. Chelton, et al, (NBS Report No. 5520, Oct. 1957).

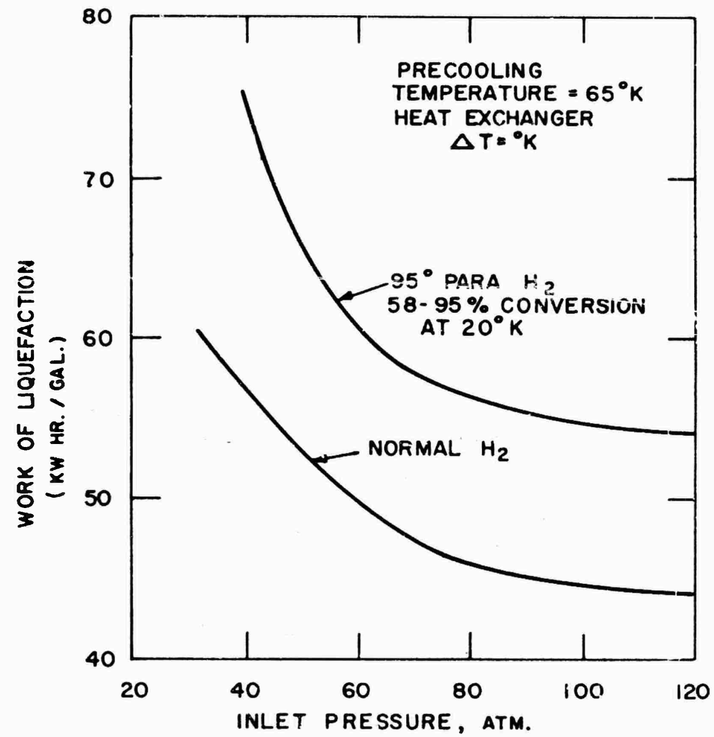

Fig. 22 - Total work of liquefaction, including precoolant, of precooled J-T H_2 liquefier, according to D. B. Chelton, et al, (NBS Tech. Note No. 5520, 1957).

A hydrogen liquefier using a precooled J-T neon cycle has been built and described by Hood and Grilly*. In this system, hydrogen was condensed at 6.6 atm. by means of liquid neon produced in a precooled J-T liquefier. The compound liquefier produced and consumed 17 liters/hr of liquid neon and produced 27.5 liters/hr of liquid hydrogen. Operating data indicate that the same apparatus produced 6 to 20 percent more liquid hydrogen than when hydrogen was used directly as the working fluid in the low temperature J-T circuit. It is also pointed out that there are no hazards associated with compressing neon.

The J-T helium liquefier may use a closed-cycle hydrogen refrigerator as a precooling device. For such a system, as shown by Chelton et al.** the work of helium compression is shown in Fig. 23 as a function of the pressure upstream of the J-T valve for precooling temperatures of 21°K and 15°K. At an assumed inlet pressure of 30 atm., and precooling temperature of 21°K, the work of compression (normalized to 67% isothermal compressor efficiency and 90% combined motor and motor-drive efficiency) is 5.6 kW-hr/lb of helium liquefied. This result assumes that ΔT_2 is zero. (See Fig. 15 for location of the ΔT's in the flow diagram).

The liquid N_2 consumption in the helium circuit of this system can be obtained from Figs. 24 and 25, also from Chelton et al.,* assuming as before that ΔT_2 is zero, ΔT_1 is 1°K and ΔT_5 is 10°K. The result is 0.69 liters N_2 per liter He liquefied, and assuming a work of 0.69 kW-hr/lb for N_2 liquefaction, this gives an additional 0.475 kW-hr/lb of He liquefied.

Finally to get the power requirement in the 21°K precooling bath, we assume this is obtained by a precooled J-T H_2 refrigerating system, as described above, with a coefficient of performance of 0.015. The ratio of the rate of helium liquefied (in liters/hr) to the 21°K refrigerator requirement in watts is given in Fig. 26 (taken from Chelton et al.*). For ΔT_1 equal to 1°K and ΔT_2 equal to zero, one obtains 0.0665 l/hr per watt, which yields a work of 3.70 kW-hr/lb of He liquefied.

Adding all the contributions, one obtains a value of 9.77 kW-hr/lb He liquefied for a precooling temperature of 21°K.

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*Hood, C.B. and Grilly, E. R. Rev. Sci. Instr. 23.357. 1952.
**Chelton, Dean, Strobridge, Birmingham and Mann. NBS Tech. Note No. 39. 1960.

V.3 SYSTEMS USING EXPANSION ENGINES

3.1 SYSTEMS WITH ONE EXPANDER FOR REFRIGERATION AND LIQUEFACTION DOWN TO TEMPERATURES OF 77°K

3.11 REFRIGERATORS FOR RELIQUEFACTION

The cycle shown in Fig. 27 is suitable for providing refrigeration over a wide range of temperatures. Refrigerators built with this cycle may use, for example, helium as the refrigerant. In this cycle, the compressed gas is first cooled in the after-cooler and then cooled by the low-pressure exhaust gas in the main heat exchanger. It is cooled by the expansion in the expansion engine and then cools the load by taking up sensible heat. After being warmed in its return path up the main heat exchanger, the low-pressure gas is recompressed, thus completing the cycle.

Early refrigerators used air as the working fluid in this cycle which has been referred to as the Allen dense-air system and the Brayton cycle. The thermodynamic diagram of the path taken by the working substance on a T-S plot is given in Fig. 28 where the lettered points correspond to those given in Fig. 27.

The essential difference between this cycle and the Stirling cycle is that, in the Stirling cycle (see Section V.4) expansion is carried out quasi-isothermally, while in the dense-gas cycle the expansion is carried out adiabatically. From a standpoint of power consumption, as will be seen in Section VIII, the Stirling cycle is the more efficient.

3.12 PRACTICAL MACHINES FOR RELIQUEFACTION OF O_2, F_2 AND N_2, THEIR COEFFICIENTS OF PERFORMANCE AND POWER REQUIREMENTS.

A. D. Little Inc., for example, has built refrigerators based on this cycle for the reliquefaction of evaporating oxygen.* In these machines helium has been the refrigerant and the high pressure side of the cycle is at 285-290 psig and the low pressure side is 70-75 psig. The expansion is carried out by a single reciprocating engine. Some models of these systems, according to A. D. Little, Inc., (A. D. L. Brochure "ADL Helium Refrigerator") have capacities of 500, 1200 and 2400 lbs O_2 condensed per day with power consumptions of 7.5, 17.5 and 33kW respectively. According to the above reference, their weights and dimensions are as follows: the 500 lb. unit, 2000 lbs. and 60" x 60" x 40"; the 1200 lb. unit, 4000 lbs. and 64" x 65" x 65" and the 2400 lb. unit, 6000 lbs. and 65" x 108" x 96".

*ADL Helium Refrigerator, Paper from Arthur D. Little, Inc.

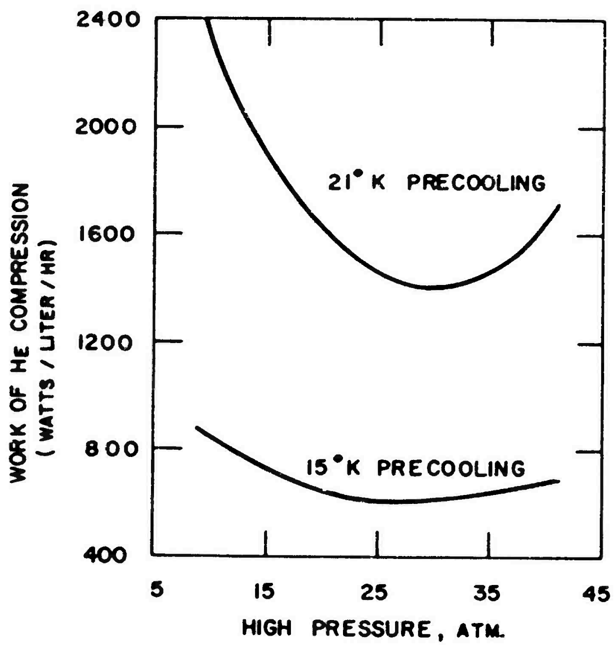

Fig. 23 - Work of helium compression in precooled J-T helium liquefier, according to D. B. Chelton, et al, (NBS Tech. Note No. 39, 1960).

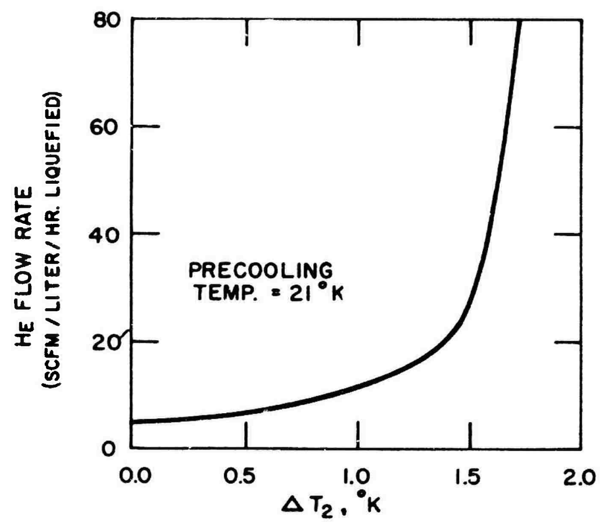

Fig. 24 - Helium flow rate in precooled J-T helium liquefier; 21°K precooling, according to D. B. Chelton, et al, (NBS Tech. Note No. 39, 1960).

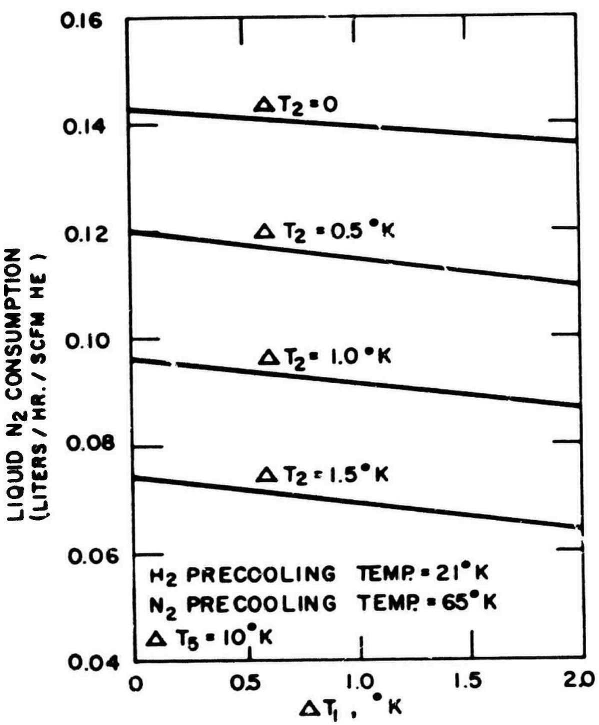

Fig. 25 - Liquid nitrogen consumption in precooled J-T helium liquefier; 21°K precooling, according to D. B. Chelton, et al, (NBS Tech. Note No. 39, 1960).

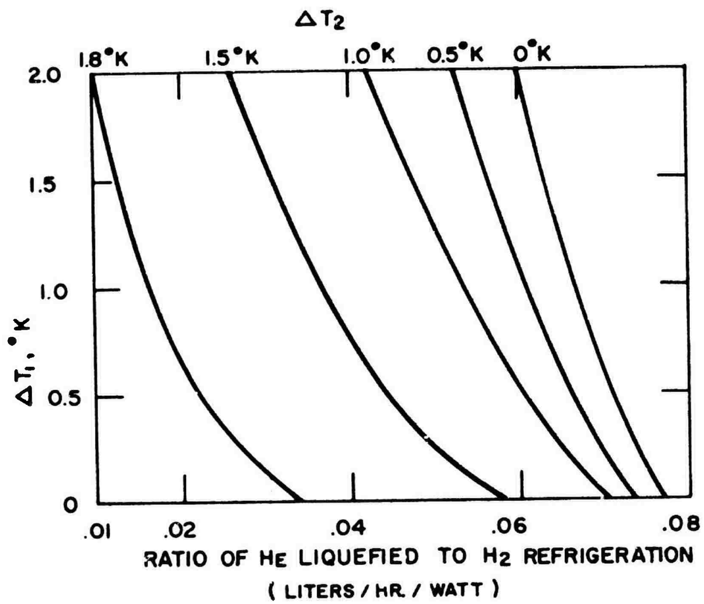

Fig. 26 - Helium liquefier-effect of heat exchanger performance; 21°K precooling. (From D.B. Chelton, et al, NBS Tech. Note No. 39, 1960).

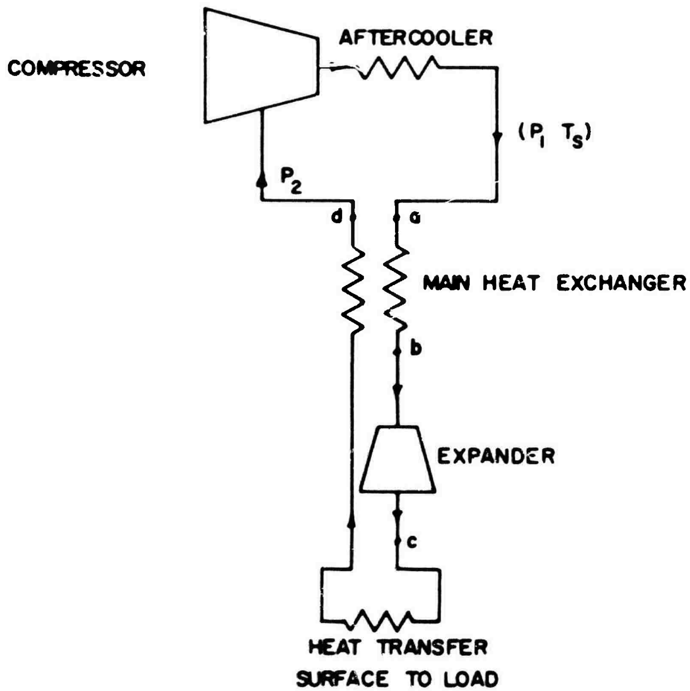

Fig. 27 - Flow diagram for closed-cycle gas refrigerator using one expander.

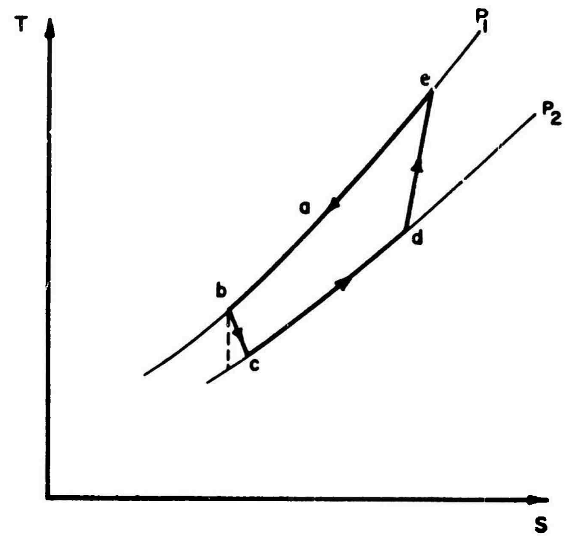

Fig. 28 - T-S diagram for closed-cycle gas refrigerator using one expander.

The actual coefficients of performance, C.P. (actual), at a lower level (7 kW input) and a higher level (15 kW input), for these machines, have been calculated from the data quoted above and from other data* and are presented in Table 12. This table also gives C.P. (Carnot) for operation at the same temperature levels, namely 77°K (oxygen reliquefaction), 85°K (fluorine reliquefaction) and 90°K (oxygen reliquefaction). The sink temperature, T_s, is taken to be 300°K.

Fig. 29 shows the power requirements, P, for various loads, L, for refrigeration at 77°K, 85°K and 90°K taking T_s = 300°K, also obtained from Table 12. In Fig. 29 the study limits for the refrigerative loads are also shown.

TABLE 12

COEFFICIENTS OF PERFORMANCE OF SOME REFRIGERATORS WITH ONE EXPANDER

(See Text for References)

T °K	C.P. (Carnot)	C.P. (Actual) (7 kW input power level)	C.P. (Actual) (15 kW input power level)
77	0.347	0.062	0.065
85	0.396	0.069	0.073
90	0.428	0.073	0.077

These data represent very conservative ratings and it is anticipated that with current and future improvements, particularly in the use of turbine expanders and centrifugal compressors that marked improvements up to about 25% are possible.

3.13 LIQUEFACTION OF O_2, F_2 AND N_2 USING MACHINES WITH ONE EXPANDER.

An open-cycle liquefaction system, analogous to the dense gas refrigerator in which the gas liquefied was the same as the working gas, would involve forming liquid in the expansion engine or turbine. While it is possible to form liquid in an expansion turbine, to the author's best knowledge there are no existing open-cycle liquefaction systems in which this is the means employed. The use of the dense gas cycle for liquefaction

*Bulletin No. HRF-1645, A. D. Little, Inc.

of O_2, F_2 and N_2 would therefore be confined to liquefaction of the desired gas in a secondary circuit in which the gas, at approximately atmospheric pressure, was permitted to condense in a vessel in thermal contact with the refrigerator evaporator.

For a single expander dense gas refrigerating machine having the same coefficients of performance as set out in Table 12, the power requirements for liquefaction are set out in Table 13, for a sink temperature, T_s, of 300°K.

TABLE 13

POWER REQUIREMENTS FOR LIQUEFACTION OF O_2, N_2 AND F_2 USING MACHINES WITH ONE EXPANDER

(For sources, see text)

Substance	ΔH 300°K to B.P. joules/g	ℓ (latent heat) joules/g	Power Requirements: Lower Power Level (7 kW input power) kW-hr/lb.	Power Requirements: Higher Power Level (15 kW input power) kW-hr/lb.
N_2	236	199	0.88	0.84
F_2	173	172	0.63	0.59
O_2	192	213	0.69	0.66

And the power requirements, based on the above data in Table 13, as a function of the rate liquefaction in liters/hr for O_2, F_2 and N_2 are shown in Fig. 30, along with the study limits. Again it should be emphasized that these figures are very conservative and marked improvements can be expected in the period 1965-70.

For nitrogen liquefaction, where it would be possible to use the same gas as the working fluid without hazards or other practical difficulties, it would be also possible to employ the Claude cycle or the Kapitza cycle, both of which come under the category of liquefiers using a single expander. Consideration of these cycles follows.

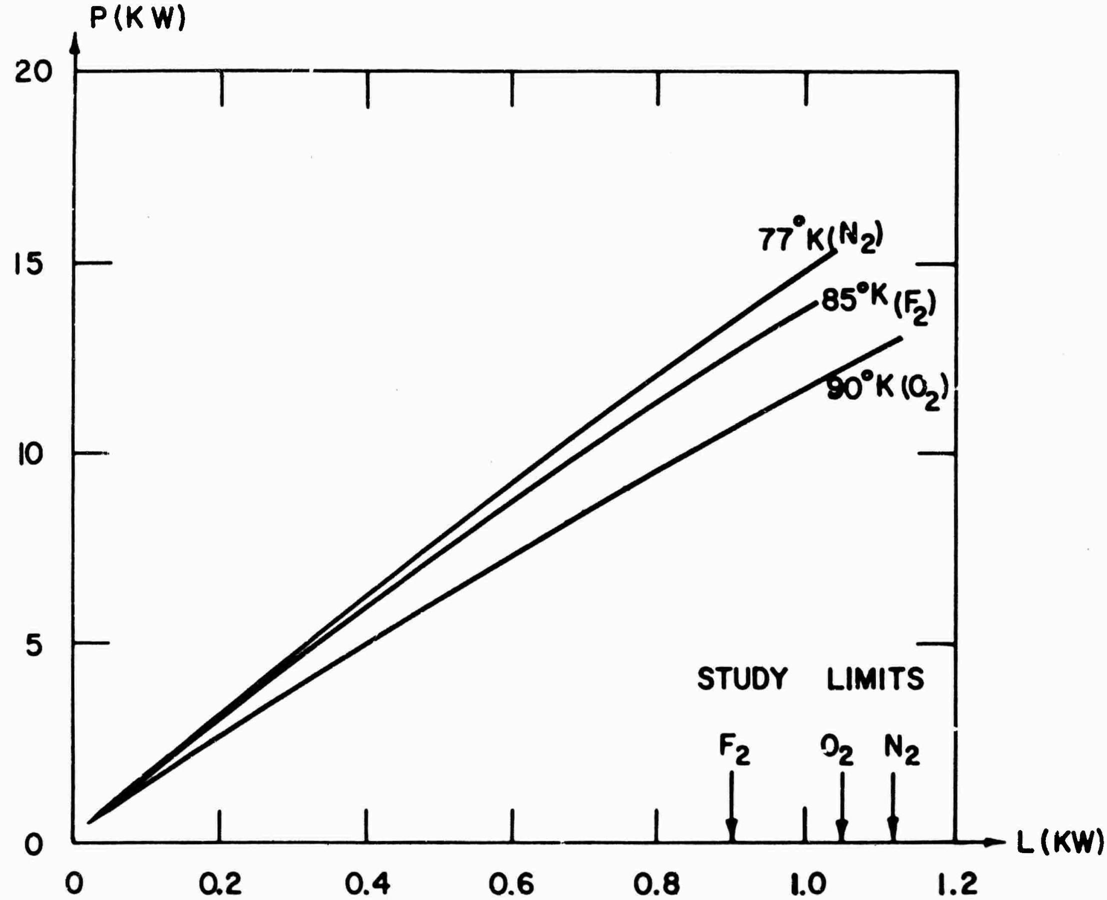

Fig. 29 - Performance of closed-cycle refrigerator with one expander using He. (From data given by A. D. Little, Inc., brochure "ADL Helium Refrigerator," see text for references.) T_s approx. 300°K. The power requirements (kW) are plotted against refrigerative load, L(kW).

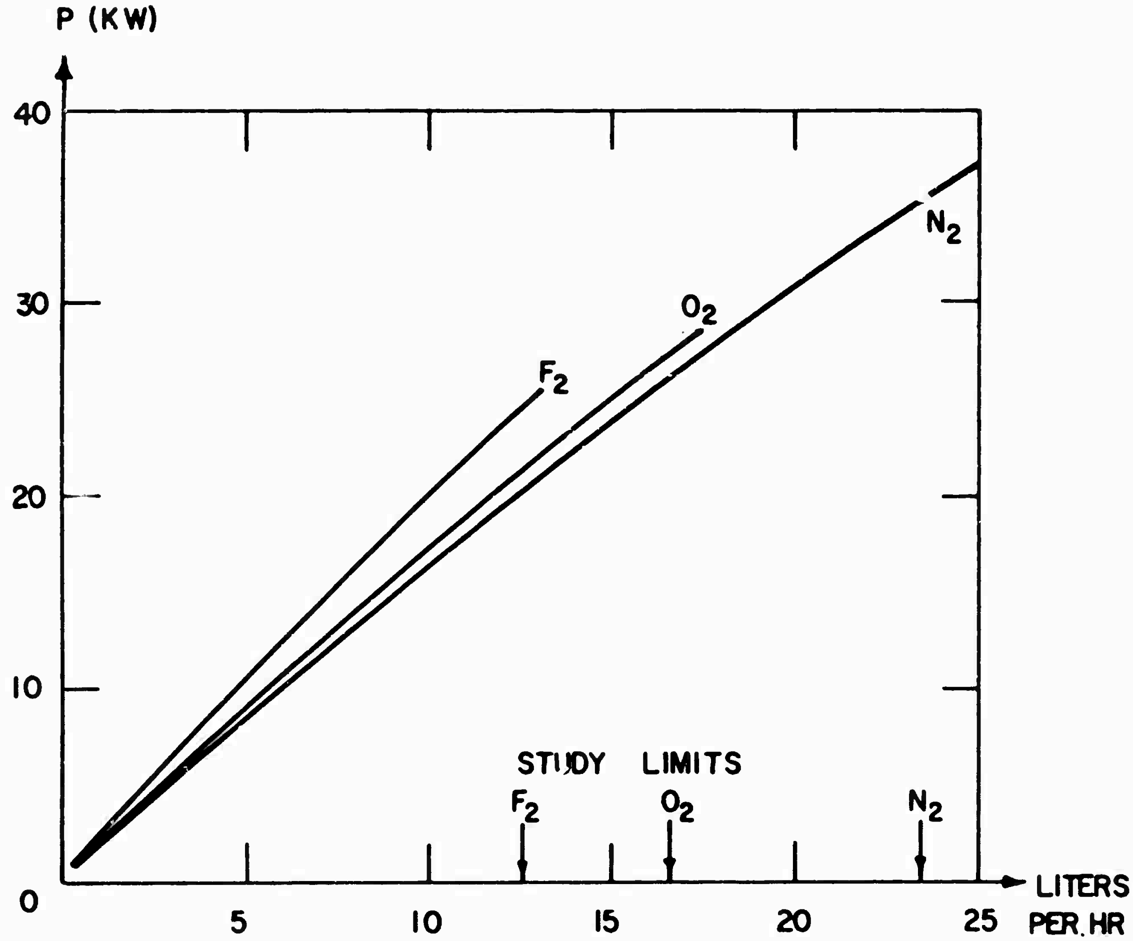

Fig. 30 - Power requirements (kW) for liquefaction of N_2, F_2 and O_2, using closed-cycle refrigerators with one expander (He gas), using data from ADL Helium Refrigerator Brochure (see text for references) versus the rate of liquefaction, (liters/hr.).

3.14 CLAUDE CYCLE

One of the earliest uses of expansion engines in gas liquefaction was in the Claude cycle, shown schematically in Fig. 31. Basically, the expansion engine serves the same purpose in this cycle as precooling in the precooled Linde liquefaction cycle. A prime source of inefficiency in the basic Linde cycle is the inability of the low pressure stream, because of its low heat capacity, completely to cool the high pressure stream. In this case, extra cooling is applied to the high pressure stream by the expansion engine. This cycle is shown on a T-S diagram in Fig. 32, in which the lettered points correspond to those in Fig. 31. It should be noted that the dotted line c-p represents perfectly isentropic expander operation, while the solid line c-f represents anticipated actual expander operation. (For simplicity in Fig. 32 the compression is shown as isothermal, along path a-b.)

Typical operational conditions for a small machine* are as follows for practical operation of a Claude liquefier, in this case for air liquefaction. (The subscripts refer to points on the flow diagram of Fig. 31.) They would be very similar for N_2 liquefaction, except that the liquefaction temperature would be 77°K.

High pressure, p_1 = 40 atm. Initial temperature, T_b 300°K. Low pressure, p_2 = 1 atm. Liquefaction temperature, T = 82°K. Temperature, T_c, of expansion engine intake = 200°K. Temperature, T_f, of expansion engine exhaust = 110°K. Temperature, T_a, at exit of first heat exchanger, E_1, = 288°K. Temperature of low pressure gas entering first heat exchanger, E_1, = 170°K, and x = 0.30.

It can be shown** that the liquefaction coefficient, ϵ, is given by:

$$\epsilon = \frac{(h_a - h_b) + (1 - x)(h_c - h_f)}{h_a - h_1} \qquad (26)$$

and under the conditions stated above this leads to a liquefaction coefficient for air of 0.14 and the coefficient of performance, assuming an isothermal compression efficiency of 67%, would be approximately 0.5 kW-hr/lb.

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*Davies, M. "The Physical Principles of Gas Liquefaction and Low Temperature Rectification," Publ. Longmans, Green and Co., 1949.
**Daunt, J.G. Hdb. d. Physik 14, 1 (1956).

TABLE 14

OPERATING CONDITIONS FOR THE CLAUDE-CYCLE AIR LIQUEFIER AT MAXIMUM EFFICIENCY

(from M. Davies, "The Physical Principles of Gas Liquefaction and Low Temperature Rectification," Publ. Longmans, Green and Company, 1949)

High pressure, p_1 (atm.)	6.5*	20	30	40	60
Fraction of gas passing through expander		0.88	0.84	0.80	0.75
Temperature of gas entering expander (°K)	115	115	173	191	215
Power required (practical) (kW-hr/lb.)	0.73	0.6	0.55	0.51	0.47

*Kapitza Machine

Table 14 shows conditions for operation of the Claude-type air liquefiers at maximum efficiency. This table also gives the coefficients of performance for air liquefaction in kW-hr/lb. for machines operating at about 25 liters/hour. We have calculated the actual practical corresponding coefficients of performance for N_2 liquefaction at a rate of approximately 1000 lbs/day on the basis outlined above and the results are shown in Table 15 below.

TABLE 15

N_2 LIQUEFACTION USING CLAUDE CYCLE. ACTUAL POWER REQUIREMENTS

Method	kW-hr/lb.
Claude Cycle (p_1 = 40 atmos.)	0.69
Kapitza Cycle (p_1 = 6.5 atmos.)	0.95

The Kapitza machine referred to in Table 15 operates on a somewhat different cycle, as shown in Fig. 33. Basically, the difference amounts to the omission of heat interchanger, E_3, from the basic Claude cycle shown in Fig. 31.

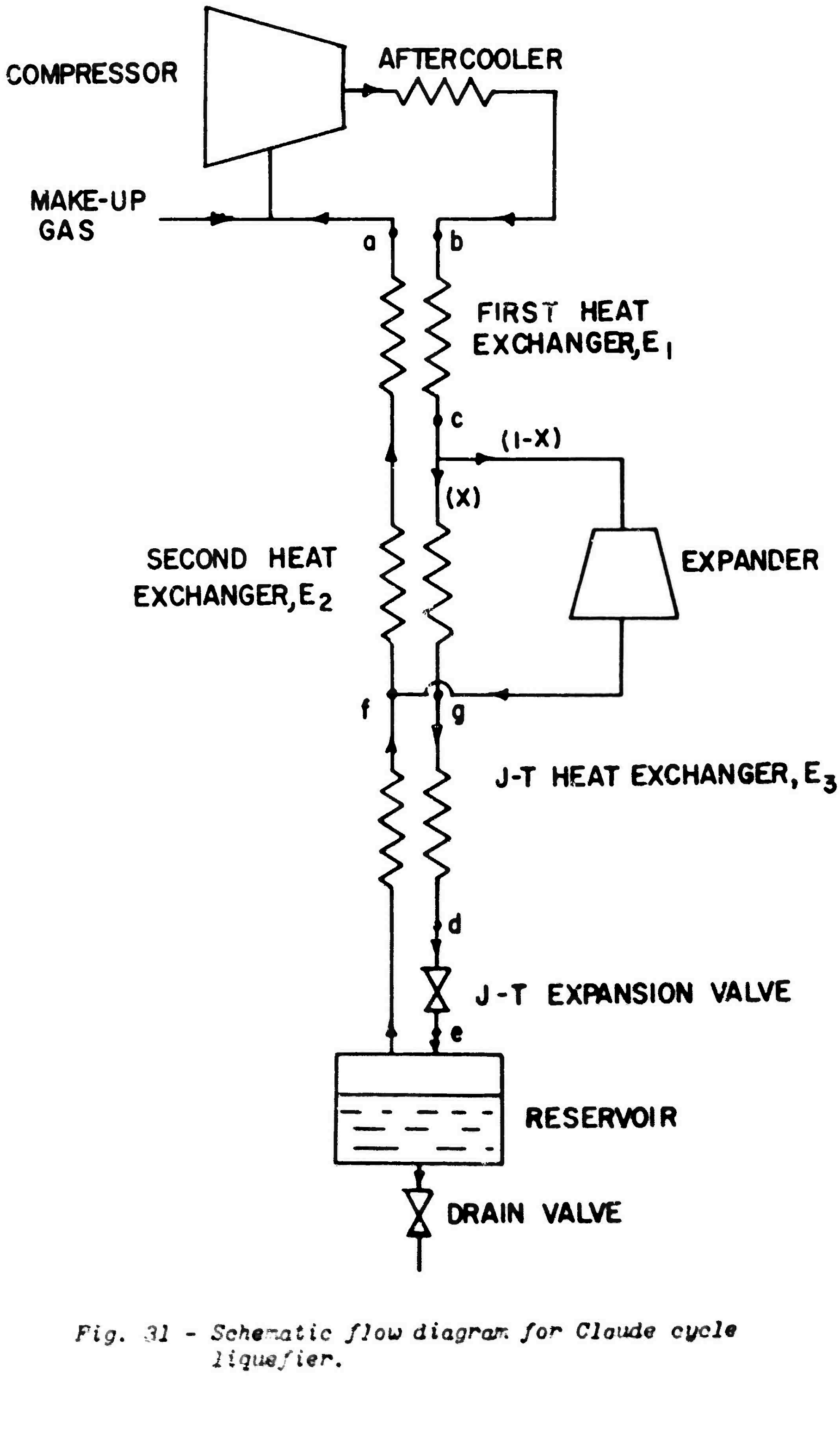

Fig. 31 - Schematic flow diagram for Claude cycle liquefier.

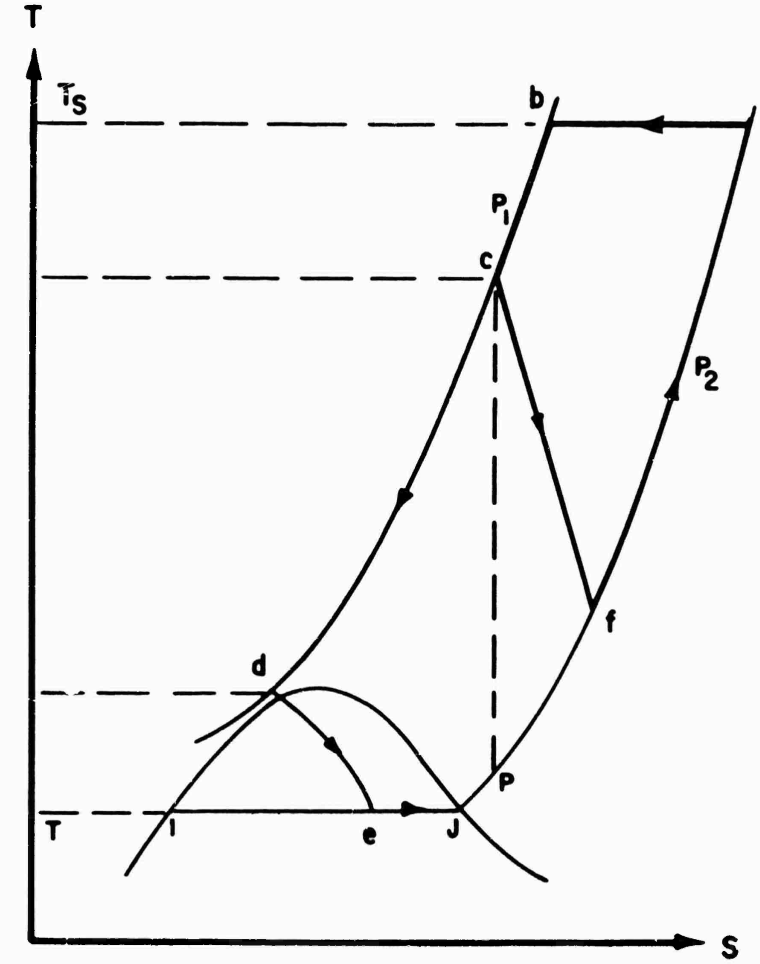

Fig. 32 - T-S diagram for Cla..de cycle liquefier.

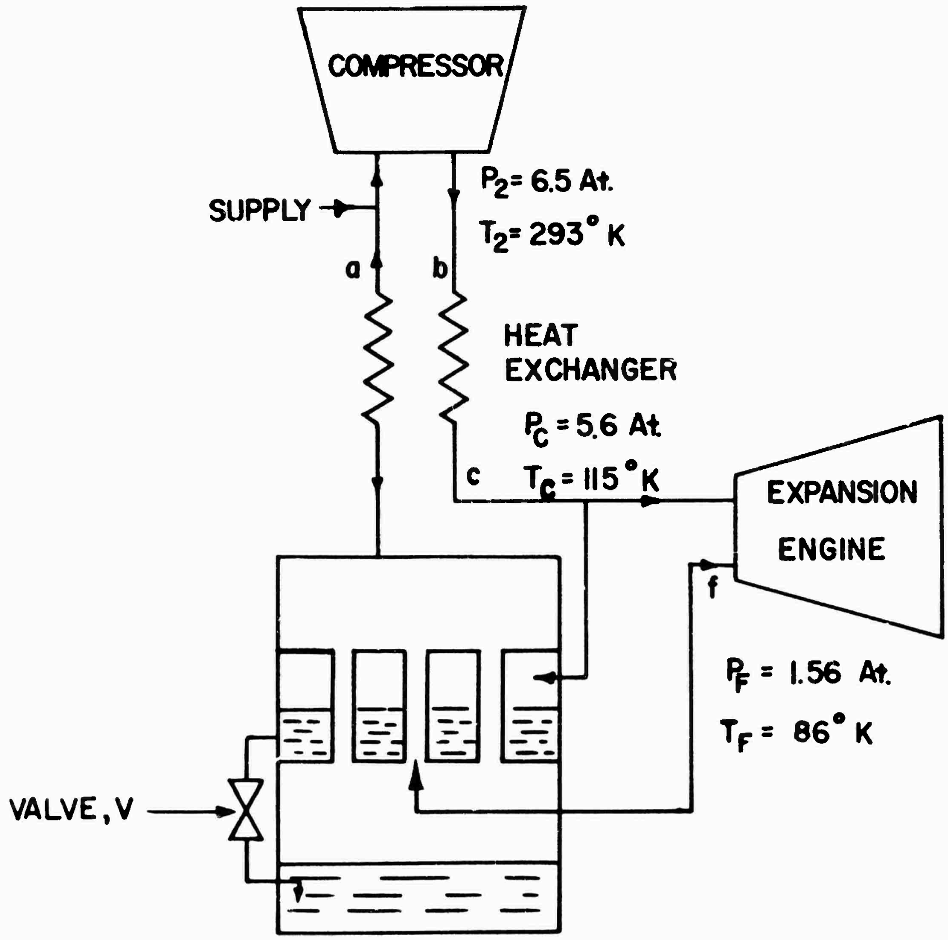

Fig. 33 - Flow diagram for Kapitza's air liquefier, using one expander.

3.2 SYSTEMS WITH ONE EXPANDER FOR REFRIGERATION DOWN TO 20°K

These systems are identical with those described in section V.3.1 which were for higher temperature operation. The flow diagram is shown in Fig. 27 and the T-S diagram in Fig. 28.

Two examples of these systems are those made by A. D. Little Inc. (for example, Model 1600 Helium Refrigerator described in ADL Bulletin HRF 1645 for a 1.6 kW refrigerative load at 20°K) and by CryoVac Inc. (HE series for 1 KW refrigerative load at 20°K described in CryoVac Bulletin 202). Both of these machines use helium as the working gas and use reciprocating expanders. Some data on them, taken from the above sources and computed by the authors, are given in Table 16.

TABLE 16

SOME OPERATIONAL DATA ON REFRIGERATORS USING ONE EXPANDER FOR REFRIGERATION AT 20°K DUE TO A. D. LITTLE INC. AND TO CRYOVAC INC.

(For references to sources, see text)

Property	A. D. Little Inc. Model 1600	CryoVac Inc. Model HE
Refrigerative load at 20°K	1.6 kW	1.0 kW
Inlet pressure	300 psia	290 psia
Output pressure	50 psia	58.8 psia
Gas flow	600 lbs/hr	350 lbs/hr
Power requirement*	141 kW	73.5 kW
Coeff. of performance*	0.011	0.013

*as computed by the authors.

In calculating the power requirements from the manufacturer's data, the authors, in order to be consistent with previous computations in this report, have assumed the isothermal compressor efficiency to be 67% and the combined motor and motor-drive efficiency to be 90%. Taking the sink temperature to be 300°K, one obtains the total power requirements and the coefficients of

performance, CP, given in Table 16. It is to be noted that the the coefficient of performance for a Carnot refrigerator operating between the same temperatures would be 0.071.

Taking from Table 16 an average practical value of CP for refrigeration at 20°K at the 1 to 2 kW level equal to 0.012, then the total power requirement for hydrogen reliquefaction at a rate of 1000 lbs/day would be 198 kW. (or 0.98 hp per liter/hr reliquefied).

The above figures are conservative ratings. It is anticipated that by the use of centrifugal expanders and compressors considerable reductions in the power requirements could be made with such single expander systems, reductions estimated to be up to 20%.

3.3 REFRIGERATION SYSTEMS WITH MORE THAN ONE EXPANDER

Trepp* has shown that, when obtaining refrigeration from expansion engines, it is thermodynamically desirable to use several expanders in series rather than one expander with a large ratio. Trepp compares cycles by evaluating the relative loss of energy, given by the expression:

$$dL = T_o \cdot dS \qquad (27)$$

where dL is the energy loss, T_o the ambient temperature and dS the entropy change.

Assuming an ideal gas and assuming utilitization of mechanical work, the multistage expansion loss (equal ratio per stage) compared with the single stage expansion loss is given by:

$$\frac{L_{single}}{L_{multi}} = \frac{\ln\left[\eta + (1-\eta)\ (P_1/P_2)^{(\gamma-1)/\gamma}\right]}{n\ln\left[\eta + (1-\eta)\ (P_1/P_2)^{(\gamma-1)\gamma^n}\right]} \qquad (28)$$

where η is the adiabatic efficiency of each expander, p_1 and p_2 the inlet and exhaust pressures, γ the ratio of the specific heats and n the number of stages in the multistage system.

A second benefit of multiple expansion is to decrease the irreversibility in the main heat exchanger of a refrigerative system. In this heat exchanger there is an interchange between the high and low pressure gas. Since the specific heat of the high pressure stream is higher than that of the low pressure

*Trepp, "Refrigeration systems for temperatures below 25°K with turbo-expanders". Adv. Cryo. Eng. 7.251.1962.

stream, it is desirable to provide additional cooling for the high pressure stream by removing it from the heat exchanger at an intermediate point, reducing its pressure and temperature in an adiabatic expander, and returning it to the exchanger to be further cooled. There is thus a smaller mean ΔT in the heat exchanger and, consequently, a smaller energy loss in it. This is illustrated graphically in Fig. 34. Fig. 35 shows a flow sheet for a dense gas cycle modified by adding intermediate expansion of the high pressure stream.

To the author's best knowledge, no refrigeration systems have been built in which multistage expansion is the sole means of cooling. The multistage expansion principle is utilized in conjunction with other means of cooling, particularly Joule-Thomson cooling. Such schemes will be discussed below in the section on compound systems.

3.4 COMPOUND SYSTEMS - GENERAL DISCUSSION

Several systems exist or have been proposed in which cooling is provided both by expansion engines and by other means, such as Joule-Thomson expansion or precooling by an external source, which is either a separate closed-cycle system or a stored liquid. These systems we shall refer to as "compound" systems. In general, expansion through engines may be substituted for expansion through valves with consequent saving of power in any of the systems previously discussed in this report. However, it is not common to use engines for expansion into the liquid region. Furthermore, in the region of vapor near the critical point an expansion engine has little thermodynamic advantage over a valve.

3.41 HYDROGEN TEMPERATURE REFRIGERATION WITH COMPOUND SYSTEMS

Trepp* describes a closed-cycle refrigerator using hydrogen as the working fluid. The cycle utilizes three stages of expansion through engines and finally a Joule-Thomson expansion forming liquid in the evaporator. The existing machine is somewhat larger than sizes of interest in this study, delivering 10 kW of refrigeration at 23°K with a coefficient of performance of 0.019. However, for smaller machines the coefficient of performance should be about the same as that measured for this larger system.

*See reference previously cited.

3.42 HYDROGEN LIQUEFACTION USING THE CLAUDE CYCLE

The Claude cycle can be used for liquefaction of hydrogen. Although it would not seem absolutely essential to employ precooling, one scheme cited in the literature does use nitrogen precooling of the high pressure hydrogen stream. The flow diagram for this scheme is shown ' "'g. 36. Chelton, Macinko and Dean* have calculated the work for liquefaction performed on the hydrogen as a function of expansion inlet temperature and pressure. The results are shown in Fig. 37. Calculations were based on an engine efficiency of 80%. Compressor efficiency was assumed to be 65% of isothermal** and the sink temperature 300°K. The total work of liquefaction is then the sum of that shown on the graphs of Fig. 37 plus that required to liquefy nitrogen. Fig. 38 shows the total work of liquefaction, assuming 65°K precooling, T_S equal to 300°K and 0.284 kW-hr/lb to liquefy nitrogen at 1 atm. For an inlet pressure of 40 atm. therefore, Fig. 38 shows that the work required is 5.35 kW-hr/lb for liquefaction of normal hydrogen and 6.50 kW-hr/lb for liquefaction and conversion to 95% para-hydrogen. (Note these values, as pointed out in the footnote, have been computed assuming an isothermal compressor efficiency of 67% and a combined motor and motor-drive efficiency of 90%.

The evaluations above for the number of kW-hr/lb, are, as noted, based on a very low figure of 0.284 kW-hr/lb for N_2 liquefaction, which can only be obtained in very large scale (tonage) plants. For operation in a space vehicle, any stream using N_2 precooling, such as the one described in this subsection or in subsection 3.43, for liquefaction up to 1000 lbs/day of H_2, would be much less efficient in the production of the required relatively small quantities of liquid nitrogen. For such systems, therefore, it is considered appropriate to assume a value of 0.69 kW-hr/lb for N_2 liquefaction, as for example has been shown in subsection 3.14. to be appropriate for small scale production using a Claude cycle. Using this value, then the revised figure for the total work of liquefaction of H_2, using the Claude cycle, becomes 8.75 kW-hr/lb for liquefaction under the same conditions as previously assumed.

*Chelton, D. B. Macinko, J. and Dean, J. W. "Methods of hydrogen liquefaction" NBS Report No. 5520. 1957.
**To make the data presented in Figs. 37 and 38 comparable with those presented earlier, the results should be modified to correspond to an isothermal compressor efficiency of 67% and then further allowance made for the combined motor and motor-drive efficiency.

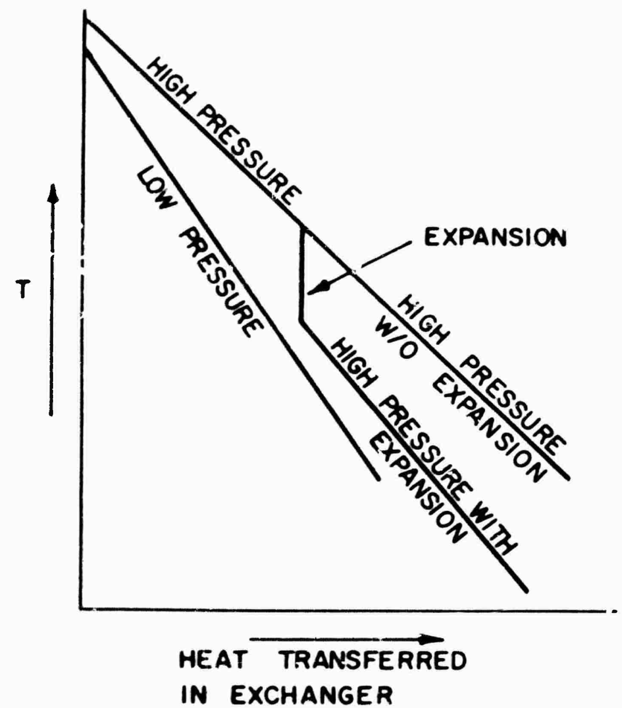

Fig. 34 - Effect of expansion engine on heat exchanger ΔT.

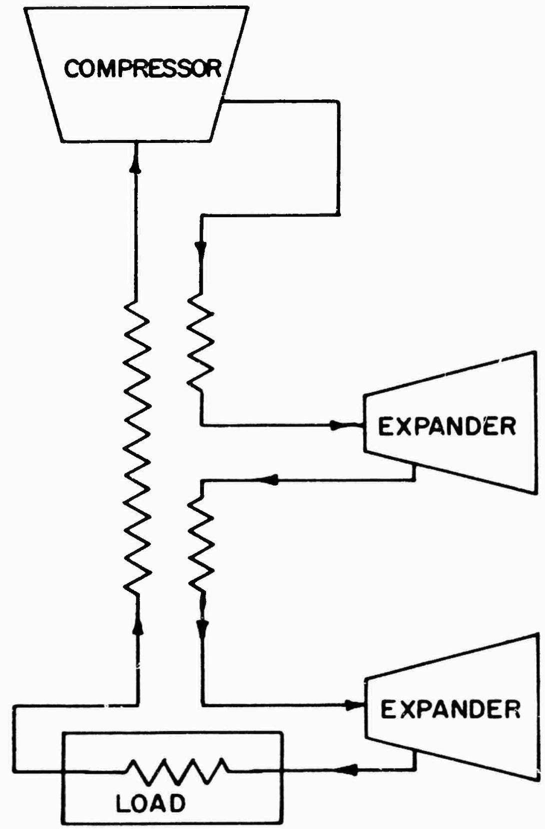

Fig. 35. Dense gas refrigeration cycle with intermediate expansion.

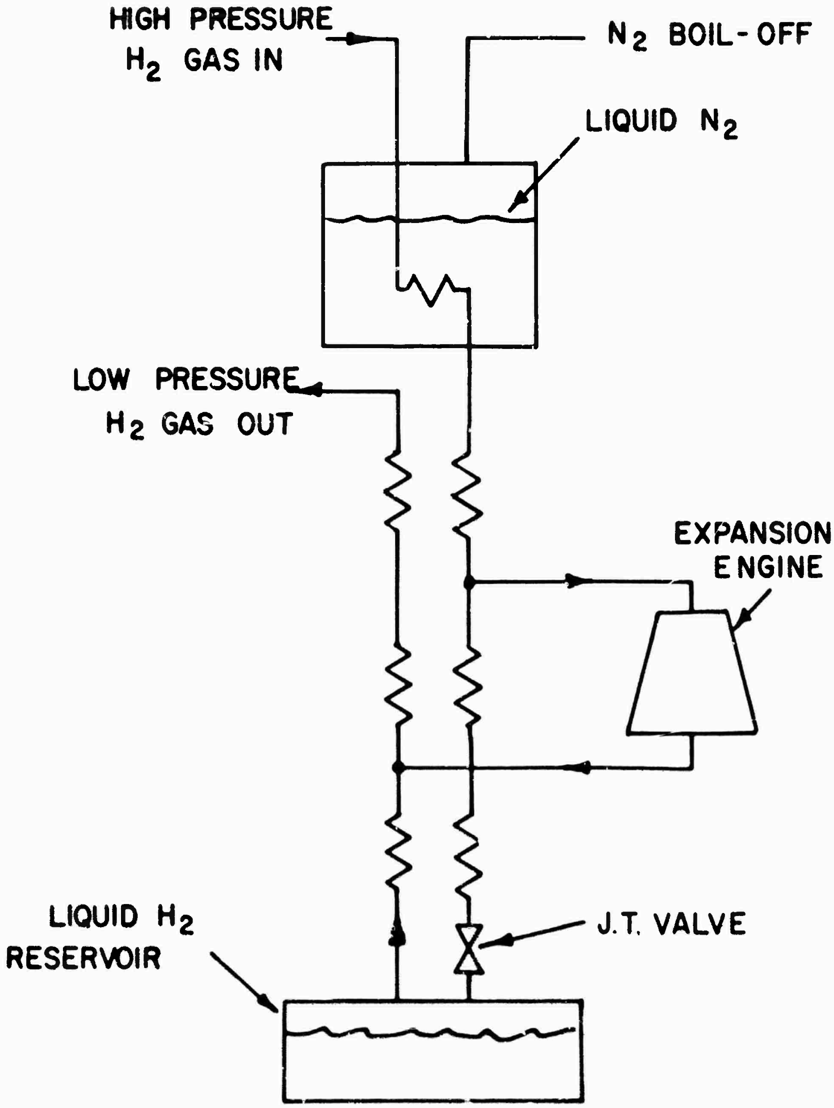

Fig. 36 - Flow diagram for hydrogen liquefier using one expander.

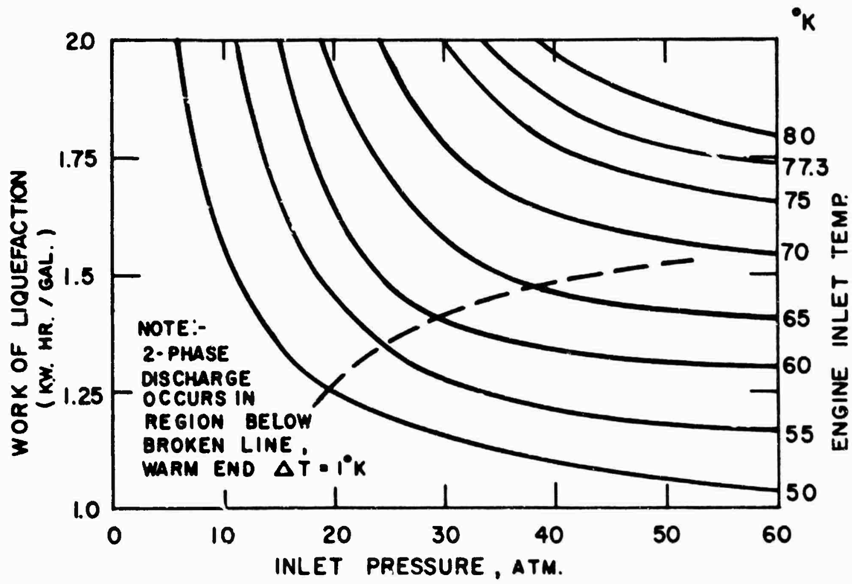

Fig. 37 - Work for H_2 liquefaction as a function of expander engine inlet temperature and pressure (excluding work required for liquefaction of precoolant N_2), according to Chelton, Macinko and Dean, (NBS Report No. 5520, 1957).

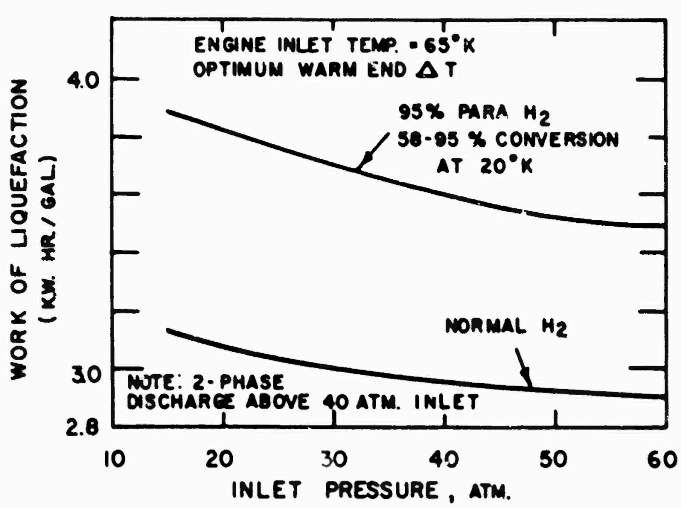

Fig. 38 - Total work of liquefaction for H_2 liquefier using one expander and liquid N_2 precoolant, according to Chelton, Macinko and Dean, (NBS Report 5520, 1957).

3.43 DUAL PRESSURE CYCLE WITH EXPANSION ENGINE FOR HYDROGEN LIQUEFACTION

Chelton, Macinko and Dean* have considered this cycle for liquefaction of hydrogen. It is identical to a two-stage J-T cycle with the exception of the substitution of an expansion engine for a valve for the high pressure expansion. The flow sheet and the T-S diagram for the cycle are shown in Fig. 39. The work of liquefaction at optimum intermediate temperatures and pressures, as calculated by Chelton et al., is summarized in Figs. 40 and 41. It may be noted that the Claude cycle is inherently somewhat more efficient thermodynamically than the dual-pressure cycle with expander. However, an advantage of the dual-pressure system is that, in the event of expansion engine failure, the defunct engine could be by-passed with an expansion valve, with liquefaction continuing at about 70% capacity. In use of the graphs of Fig. 41, note again must be taken, as discussed above in subsection 3.42, of the small value in kW-hr/lb for the liquefaction of the N_2 precoolant assumed by Chelton. et al. If, as assumed previously, the N_2 liquefaction on a small scale takes 0.69 kW-hr/lb, then this system requires total work of 9.81 kW-hr/lb for H_2 liquefaction with 80 atm. H_2 inlet pressure.

3.44 MULTISTAGE EXPANSION WITH LIQUEFACTION IN JOULE-THOMSON VALVE FOR HELIUM LIQUEFACTION

This type of process is common in the liquefaction of helium, as exemplified in the Collins cryostat, marketed by A. D. Little, Inc. One of the Collins-type helium liquefiers uses three stages of isentropic expansion followed by expansion through a valve. Such a machine has been built by S. C. Collins at the Massachusetts Institute of Technology. The temperature drops in the three expanders are roughly as follows: 80°K to 45°K; 45°K to 25°K; 15°K to 9°K. For a rate of liquefaction of 25 to 32 l/hr with liquid N_2 precooling, the power required (N_2 plant excluded) is 45 kW and the actual work expended, including that on N_2 plant, is 11.5 kW-hr/lb.** It is anticipated that for larger scale operations, as for the upper limit considered in this study of 155 l/hr, centrifugal compressors and expanders could be used with very marked improvement in performance.

An alternative derivation of the power requirement would be to combine the data for the coefficient of performance for a multi-engine hydrogen refrigerator with J-T cooling (see

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*NBS Report No. 5520. 1957

**See, for example, Scott, R. B., "Cryogenic Engineering," Publ. Van Nostrand Co., 1959.

subsection 3.41) with the data given in Section 2.5 for the performance of a J-T helium liquefier precooled to 21°K. Referring to section 2.5, the helium compressor power for the helium circuit is 5.6 kW-hr/lb of helium liquefied, to which must be added 0.48 kW-hr/lb for the N_2 precooling in the helium circuit. In addition, the 21°K refrigeration, assuming a coefficient of performance of about 0.02, (see subsection 3.41), would require 2.78 kW-hr/lb. The total therefore is 9.86 kW-hr/lb.

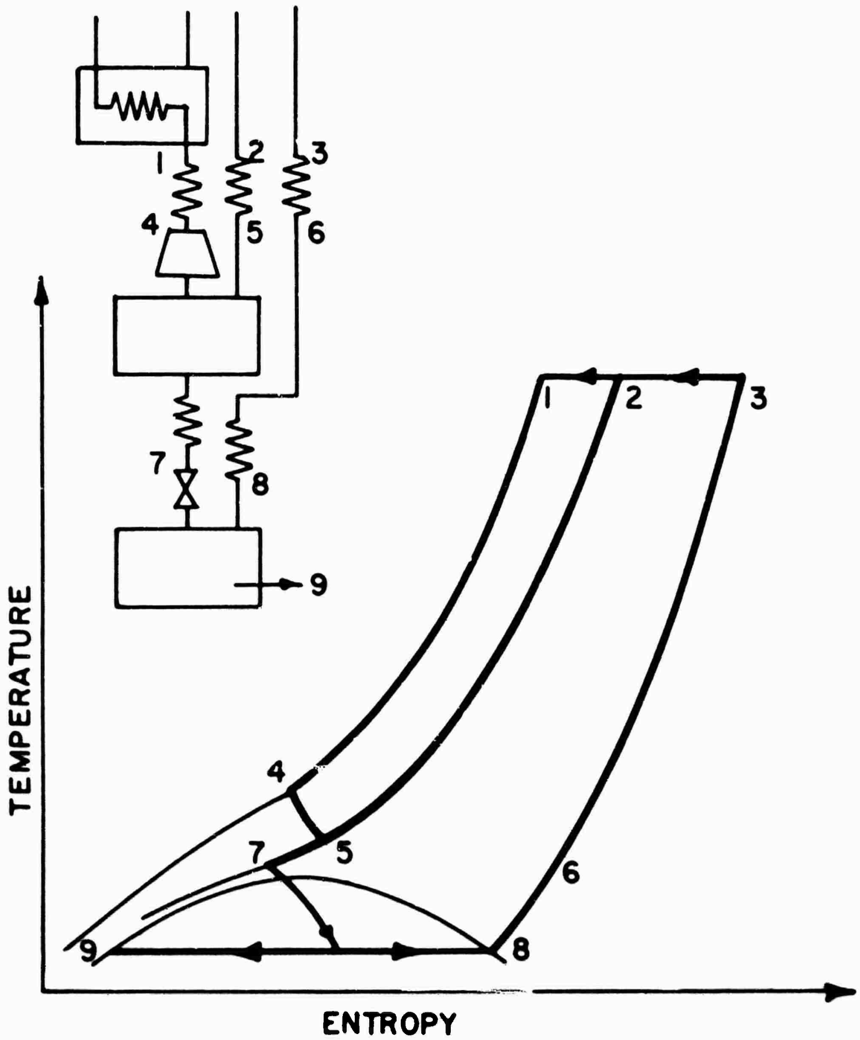

Fig. 39 - Flow diagram and T-S diagram for dual pressure cycle with expander engine for liquefaction of H_2, according to Chelton, Macinko and Dean, (NBS Report No. 5520, 1957).

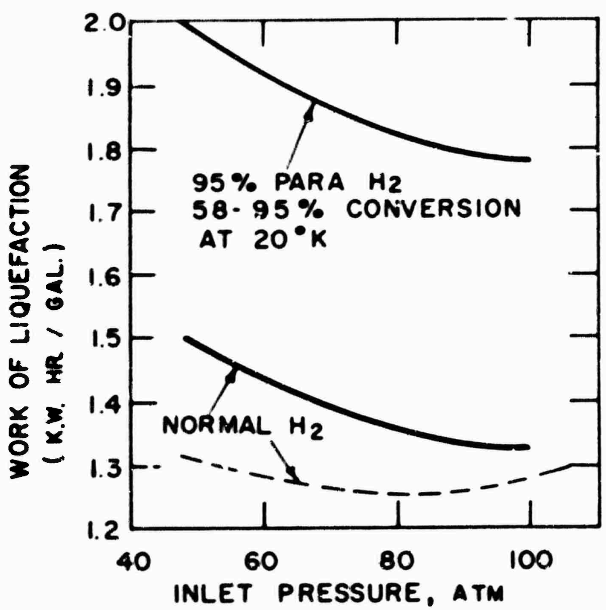

Fig. 40 - Work of liquefaction of H_2 (excluding precoolant) at optimum conditions for dual pressure H_2 liquefier using one expander, according to Chelton, Macinko and Dean, (NBS Report No. 5520, 1957).

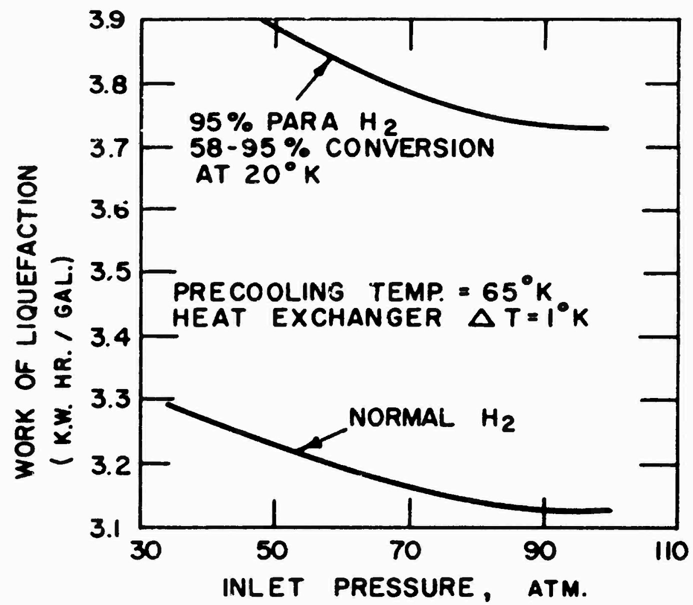

Fig. 41 - Total work of liquefaction of H_2 for dual pressure H_2 liquefier using one expander, according to Chelton, Macinko and Dean, (NBS Report No. 5520, 1957).

V.4. STIRLING CYCLE SYSTEMS

4.1 GENERAL DESCRIPTION OF CYCLE

Refrigerators, based on the Stirling cycle, have been sucessfully and efficiently developed for liquefaction down to N_2 temperatures.* Engines operating to lower temperatures, e.g., H_2 temperatures, are currently being developed.** Although this technique is not new, having been adopted for example by Kirk for ice-making machines in 1862, its method of operation is not generally familiar. A brief description of an idealized Stirling cycle refrigerating engine therefore follows. The fundamental requirements for such an engine are to have a compressor volume, V_c, at the higher temperature, T_c, and an expansion volume, V_E, at the lower temperature, T_E, and to allow a fixed quantity of gas to pass alternately from one volume to other via a regenerator, no valves being used (see Fig. 42). In order to obtain a refrigerating effect, the movements of the two pistons in the two volumes must be out of phase, such that the expansion volume leads with respect to the compressor volume by about 90°.

The necessary steps in the functioning of a refrigerating engine operating on an idealized Stirling cycle are set out in Fig. 42 and are as follows:

1. Isothermal compression of the gas in V_c with V_E zero. The heat of compression is ejected at the head of the compressor at temperature, T_c, and ideally the process can be considered isothermal. On the indicator diagram of Fig. 42 this step follows the isothermal curve at temperature T_c from point A to B.

2. Constant volume gas transfer from V_c to V_E. This step, illustrated by the path B to C along the isochoric path on the indicator diagram of Fig. 42 involves no work and is achieved by simultaneous and equal increase in V_E and decrease in V_c. During this process the gas in passing through the regenerator gives up heat so as to emerge into V_E at temperature T_E.

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*Kohler, J. W. L. and Jonkers, C. O. Philips Tech. Rev. 16.69 and 105 (1954). See also Malaker Labs., Inc., brochure "The Cryomite."
**Malaker, F. Rossi, R. and Daunt, J. G. Paper presented at tenth National Infrared Information Symposium, Fort Monmouth, N. J., October 1963

3. Isothermal expansion of the gas in V_E with V_c zero. The refrigeration produced by the expansion results in heat absorbed at the expander at temperature T_E, and ideally the process can be considered isothermal. On the indicator diagram of Fig. 42, this step follows the isothermal curve at temperature T_R from point C to D.

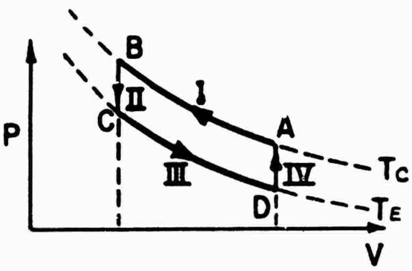
4. Constant volume gas transfer from V_E to V_c. This step, illustrated by the path D to A along the isochoric path on the indicator diagram of Fig. 42 is achieved by simultaneous and equal decrease in V_E and increase in V_c. During this process the gas in passing through the regenerator picks up the heat stored there during process 2, and it emerges in V_c at temperature T_c.

The cycle being one between two operational temperatures is, ideally, as efficient as a Carnot cycle.

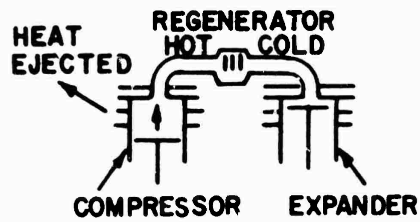
In practice the most conveniently arranged piston movements for the compressor and expander are harmonic or approximately harmonic. Such a harmonically operated system modifies the idealized Stirling cycle outlined above, but does not affect the basic principle of operation. The system, particularly in view of the complete absence of valves, has advantages of mechanical simplicity and reliability. It also lends itself to cascaded operation. A further noteworthy advantage is that it requires no external compressors.

4.2 PRACTICAL SINGLE-STAGE MACHINES

The Philips Company and Malaker Laboratories, Inc., produce modified Stirling cycle refrigerators. The former company (under Cryogenerators, Inc.) produce two reliquefiers or refrigerators capable of operating down to about 75°K with refrigerative capacities of 0.9 kW at 78°K for an input power of 11 to 11.5 kW and of 3.6 kW at 78°K for an input power of 41 to 42 kW. It also produces a miniature unit with capacity of 1 watt at 30°K for an input power of about 250 watts and of 10 watts at 80°K. The latter company produces a miniature unit, the Cryomite, with a capacity of about 8 watts at 77°K with an input power of approximately 220 watts, and are developing units for refrigeration at 20°K with capacity up to about 1 kW. In all cases the heat rejection is to ambient temperature (300°K).



I ISOTHERMAL COMPRESSION



Compressor piston moves up.
Expander piston stationary at top.
Heat given out by compressed gas.

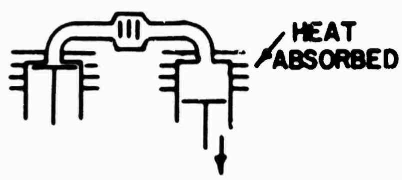
(Path A to B on indicator diagram)

II CONSTANT VOLUME GAS TRANSFER

Compressor piston moves up.
Expander piston moves down so that gas is transferred from compressor to expander at constant volume.

(Path B to C on indicator diagram)

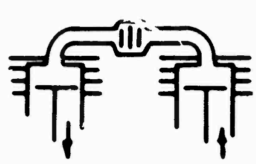
III ISOTHERMAL EXPANSION



Compressor piston stationary at top.
Expander piston moves down.
Heat taken in by expanding gas.

(Path C to D on indicator diagram)

IV CONSTANT VOLUME GAS TRANSFER



Compressor piston moves down.
Expander piston moves up so that gas is transferred from expander to compressor at constant volume.

(Path D to A on indicator diagram)

Fig. 42

Steps in the operation of an idealized refrigerator based on the Stirling cycle.

3 COEFFICIENTS OF PERFORMANCE, POWER REQUIREMENTS FOR REFRIGERATION AND LIQUEFACTION AT TEMPERATURES OF 77°K AND ABOVE

The ideal coefficients of performance, C.P., of Stirling cycle engines, even when operating with harmonical piston movements, are the same as those of a Carnot engine, namely:

$$C.P. = T/(T_S - T) \qquad (29)$$

where T is the temperature of the refrigerative load and T_S the temperature of the heat sink.

Actual coefficients of performance are in practice smaller than the Carnot values due to several causes, the most important being: (1) non-isothermal conditions in expansion and compression, (2) regenerator and heat exchanger inefficiencies, (3) heat losses due to imperfect thermal isolation of the cold parts of the engine, (4) drive motor inefficiency. It is convenient to assess the actual practical C.P. values for machines operating down to 77°K by consideration of observed values of the coefficients of performance of existing Stirling cycle engines with refrigerative load capacity similar to those for the maximum loads of interest in this report (see Section III). The practical evaluations of C.P. (actual) for actual machines have been taken from data by Kohler and Jonkers,* who gave data for the shaft power required for various refrigerative loads at different temperatures. The values given by Kohler and Jonkers have been diminished by a factor (100/85), in order to account for drive motor inefficiency. The motor efficiency has been taken to be 85% for the higher level of operation considered (1000 lbs/day). The data, so adjusted for motor inefficiency, is shown in Fig. 43, which plots C.P. (actual) as a function of the refrigerative load temperature, T, under the condition that the sink temperature T_S is 300°K. The data are applicable for operation with a refrigerative load of about 1 kW.

Estimates of the power requirement, P, for refrigeration at the three temperatures of interest, namely (1) 77°K (nitrogen reliquefaction), (2) 85°K (fluorine reliquefaction), and (3) 90°K (oxygen reliquefaction) were made in the following manner. At the higher power levels (approx. 1 kW, the C. P. values were taken from the curve given in Fig. 43. At the lower power levels (approx. 0-0.3 kW) more conservative C.P. values were adopted.** A summary of this data is given in Table 17. The power requirement, p, in kilowatts for a given refrigerative load, L, (in kW) is then given in the following equation:

*Kohler J. W. L. and Jonkers, C. O., Philips Tech. Rev. 16, 105 (19[illegible]).

**Taken from data given in Cryogenerator's Bulletin, CP1000-5M-862 and from unpublished data by Malaker Laboratories, Inc.

$$C.P. = \frac{L}{P} \tag{30}$$

TABLE 17

VALUES OF C.P. FOR STIRLING CYCLE REFRIGERATORS OPERATING DOWN TO 77°K

(For sources, see text)

T	C.P. (Carnot)	Low Power Level C.P. (actual)	Higher Power Level C.P. (actual)
77°K	0.347	0.075	0.084
85°K	0.396	0.088	0.114
90°K	0.428	0.093	0.133

The results are also shown in Fig. 44, which plots P versus L for various refrigerative loads at temperatures of 77°K, 85°K and 90°K, taking T_s = 300°K. Also in Fig. 44 are shown the ideal (Carnot) power requirements under the same conditions, and the study limits for the refrigerative loads.

In employing a Stirling cycle engine for liquefaction, the gas to be liquefied would be introduced at approximately atmospheric pressure into thermal contact with the cold head of the refrigerator in a separate gas circuit.

To estimate the power requirements for liquefaction by the Stirling cycle method, it is to be noted that the sensible heat of the gas at approximately atmospheric pressure from T_s to the boiling point acts as a full load on the refrigerator as well as the latent heat. The data for and the results of the computations are given in Table 18.

From these data and unpublished data of Malaker Laboratories, Inc., on very low power engines (miniature engines), the curves of Fig. 45 were drawn, which show the power p required for liquefaction as a function of the liquefaction rate in liters per hour for N_2, F_2 and O_2.

Finally, it is advantageous to express the liquefaction figure of merit in terms of kW-hr per lb of substance liquefied. This data is presented in Table 19 for the two power levels.

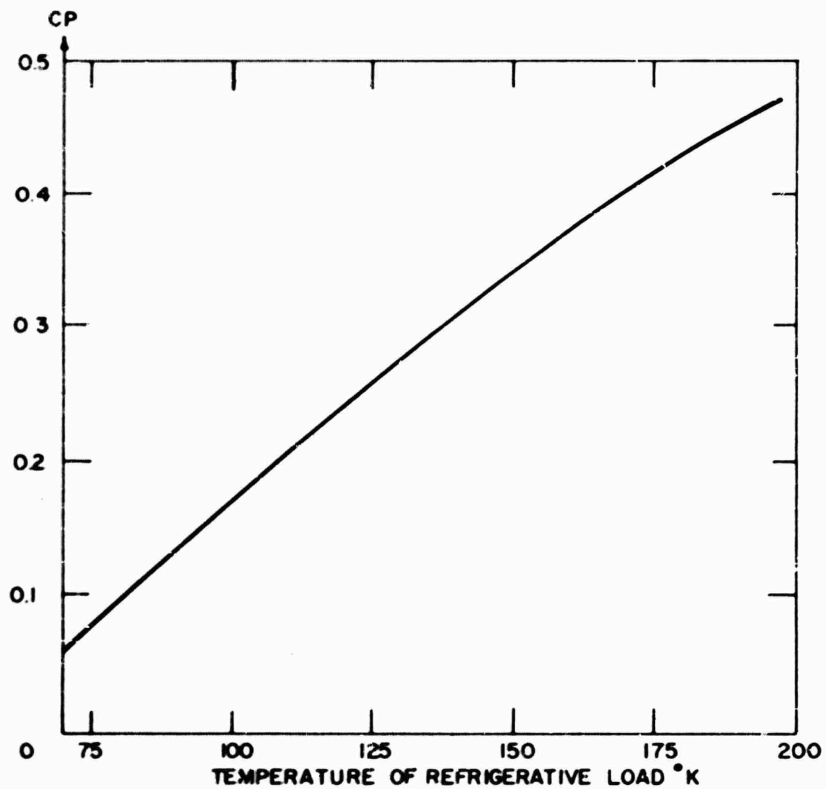


Fig. 43 - Coefficient of performance, CP, of a Stirling cycle refrigerator operating at a refrigerative load level of about one kW as a function of the temperature of the refrigerative load. (From data of Kohler and Jonkers, see text for reference, modified for motor inefficiencies). T_S taken to be 300°K.

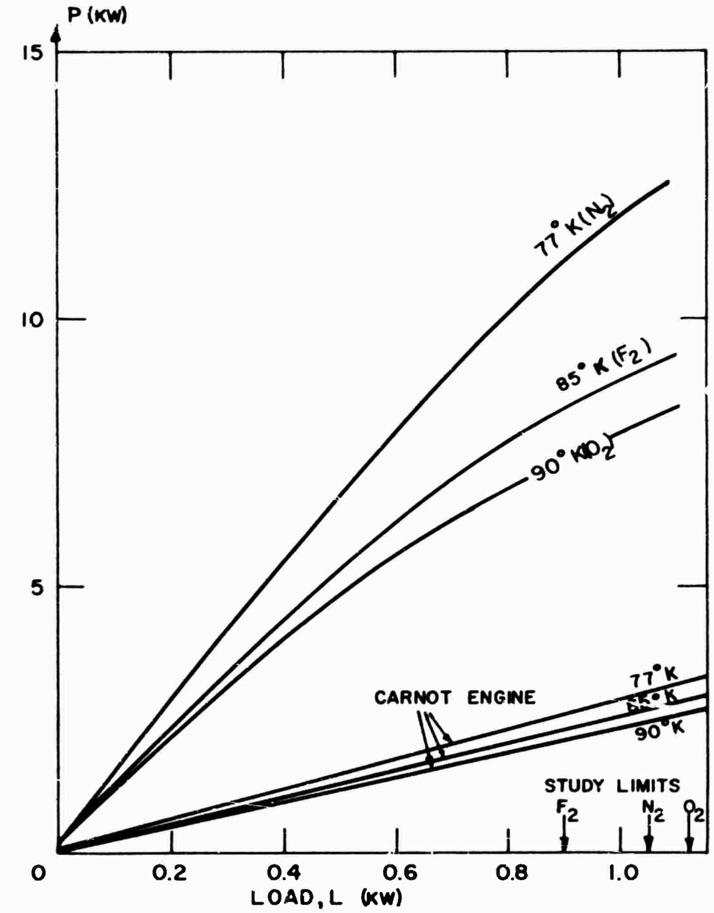


Fig. 44 - Power requirements (kW) versus refrigerative load, L(kW) for Stirling cycle refrigerators. T_S taken to be 300°K. (For source of data, see text.)

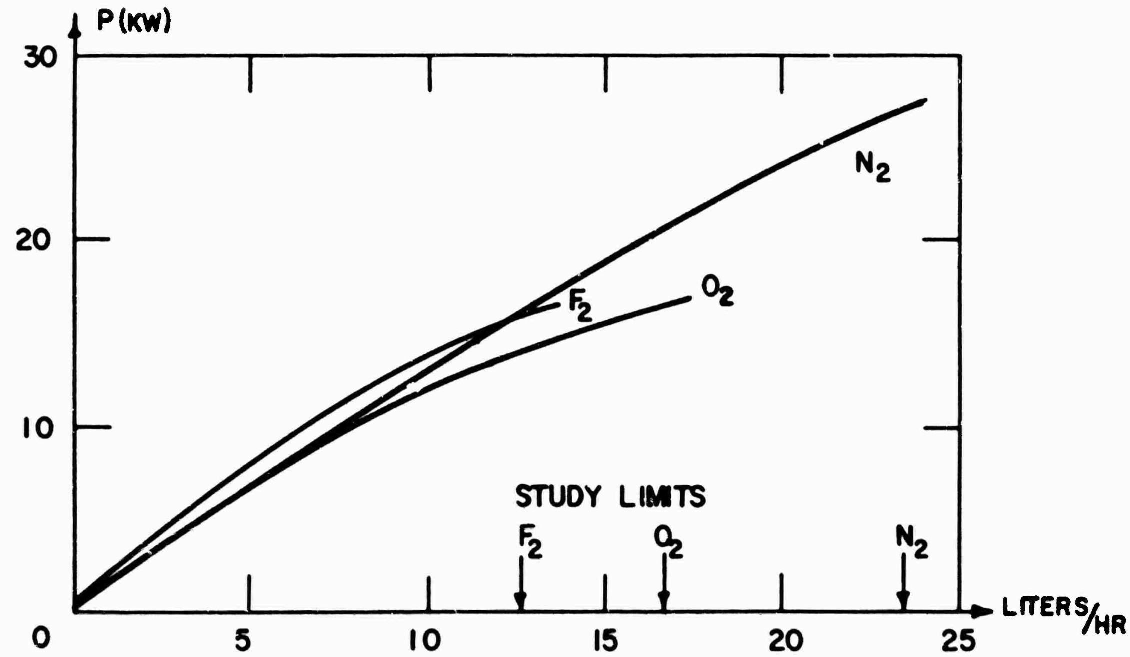


Fig. 45 - Power requirements (kW) for liquefaction of N_2, F_2 and O_2, using a single-stage Stirling cycle refrigerator system. T_3 is taken to be 300°K. (For source of data, see text.)

TABLE 18

POWER REQUIREMENTS FOR LIQUEFACTION OF N_2, O_2 AND F_2 BY STIRLING CYCLE MACHINES

Substance	ΔH 300°K to B.P. joules/g	ℓ (Latent heat) joules/g	Low Power Level		Higher Power Level	
			C.P. (actual)	P for 100 lbs/day kW	C.P. (actual)	P for 1000 lbs/day kW
N_2	236	199	0.075	3.04	0.084	27.1
F_2	173	172	0.088	2.07	0.114	15.8
O_2	192	213	0.093	2.27	0.133	16.0

TABLE 19

LIQUEFACTION FIGURE OF MERIT FOR STIRLING CYCLE MACHINES OPERATING DOWN TO 77°K

Substance	Boiling Point	Liquefaction Figure of Merit	
		Low power level 100 lb/day)	High power level (1000 lb/day)
N_2	77°K	0.73 kW-hr/lb	0.65 kW-hr/lb
F_2	85°K	0.50 kW-hr/lb	0.38 kW-hr/lb
O_2	90°K	0.55 kW-hr/lb	0.38 kW-hr/lb

All the above data are conservative ratings of the power requirements based on current practice. It is anticipated that for the period 1965-70 some improvement can be made, mainly by improvement of drive motor and drive mechanism efficiency, by upgrading regenerator and heat exchanger efficiency and by attention to heat losses. Optimistic figures for all the power requirements for Stirling cycle liquefiers and reliquefiers should be based on at least a 15% improvement on the data given above.

4.4 EFFECT OF SINK TEMPERATURE, T_s, VARIATION ON THE COEFFICIENT OF PERFORMANCE OF STIRLING CYCLE REFRIGERATORS WORKING DOWN TO 77°K.

As is discussed in Section VII.2, the sink temperature of space liquefiers and reliquefiers is determined by the temperature

of the radiator. Moreover, as will be seen in Section VII.2.5, it is desirable if possible to use high radiator temperatures in order to save radiator weight. However, higher values of T_s result in reduced values of the coefficient of performance, C.P., of refrigerators.

Based on the data for Stirling cycle refrigerators operating at about the 1 kW load level, it has been possible to estimate the effect of variation of T_s.

In the calculation, it has been assumed that for refrigeration at a given temperature level the figure of merit, η_{rel}, remains constant. By varying T_s, the Carnot coefficient of performance, C.P. (Carnot) can be readily calculated for any given values of T_s and the refrigeration temperature T. If η_{rel} remains the same, then C.P. (actual) is immediately obtained and the results should be correct in first order. The results of such computations are shown in Fig. 46 which shows C.P. (actual) for the Stirling engines described in Section 4.3, as a function of the sink temperature T_s when refrigerating at 77°K, 85°K and 90°K. It will be seen that, by going to lower T_s values, marked gains in efficiency can be made.

4.5 CASCADED STIRLING CYCLE ENGINES FOR RELIQUEFACTION AND LIQUEFACTION OF H_2 AND H_e

One significant advantage of the Stirling cycle system is that it can be readily cascaded to provide temperatures down to about 15°K. In practice such cascaded systems have been made by Malaker Laboratories Inc. and these engines involve the use of multiple expansion volumes which are provided by a single stepped piston device. Such cascaded systems can be two or three stage or can be made into compound systems in which a cascaded Stirling cycle engine serves as a precoolant for a subsidiary Joule-Thomson cycle for H_2 or He reliquefaction or liquefaction.

The following arrangements, amongst others, are under development by Malaker Laboratories Inc.: (1) Small scale single stage Stirling cycle providing precooling at 65°K for a J-T H_2 circuit to provide refrigeration at 20°K or to liquefy H_2. For this the maximum design coefficient of performance is 0.015 for refrigeration at 20°K and the minimum work for H_2 liquefaction 11 kW-hr/lb. (2) Small scale two stage cascaded Stirling cycle system to provide 30°K precooling for a J-T H_2 cycle. For this the rating for the coefficient of performance is > 0.009 for refrigeration at 20°K and the work of liquefaction of H_2 <19.5 kW-hr/lb. (3) Small scale three stage cascaded Stirling cycle system to provide 15°K precooling for a J-T He cycle for refrigeration at helium temperatures with coefficient of performance, of 0.0012 and with a design work of liquefaction of 13 kW-hr/lb He.

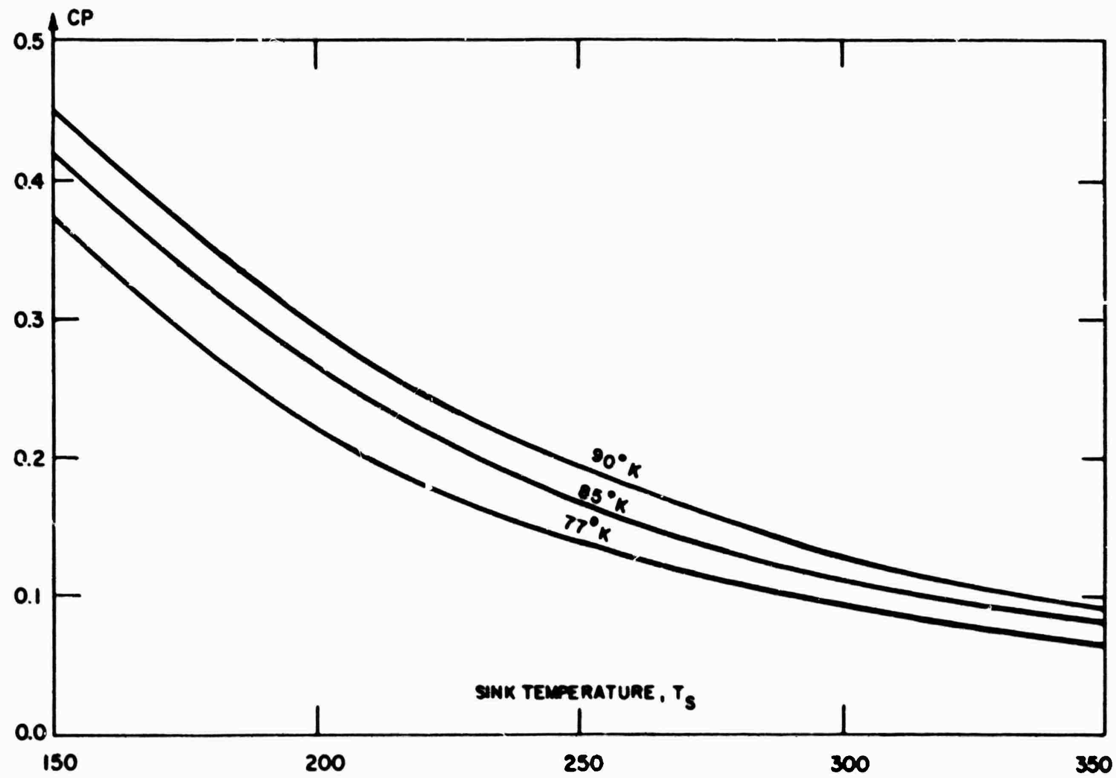


Fig. 46 - Variation of the coefficient of performance, CP, as a function of the sink temperature, T_s, for a single-stage Stirling cycle refrigerator when operating at 77°K, 85°K and 90°K. (For sources of data, see text.)

V.5 TACONIS CYCLE SYSTEMS

5.1 GENERAL DESCRIPTION OF THE TACONIS CYCLE SYSTEM

This method was suggested by Taconis* in 1950. Fig. 47 shows a diagramatic sketch of the steps followed in an idealized arrangement. The main feature of the engine is a displacement expander, D, consisting of a piston closed at both ends moving up and down in a slightly longer cylinder, C, also closed at both ends. The spaces at the top and bottom of the cyclinder are interconnected by gas passages via a regenerator, R. The top space is connected to the high pressure and low pressure sides of a compressor through the inlet and exhaust valves. The cycle of operations, which is a closed cycle, is as follows:

1. Isothermal compression. With the displacer piston at the bottom, the inlet valve is opened and the top space compressed from p_1 to p_2. The heat of compression in the top space is ejected at the head of the cylinder and ideally the process can be considered to be isothermal at temperature T_1.

2. Constant pressure gas transfer. In this step the piston is moved from the bottom to the top of the cylinder with the inlet valve remaining open. The gas is transferred via the regenerator at constant pressure to the bottom space. During this process the gas in passing through the regenerator gives up heat so as to emerge at the lower temperature, T_2

3. Isothermal expansion. With the displacer piston remaining at the top of the cylinder, the exhaust valve is opened and the gas expands from p_2 to p_1. During this process heat is absorbed at the lower end of the cylinder and ideally this process can be considered isothermal. In passing back through the regenerator, the gas receives heat from it so as to emerge at the upper end of T_1

4. Constant pressure gas transfer. Finally when the gas pressure is reduced to p_1, the displacer piston is allowed to fall to the bottom of the cylinder, gas passing back along the regenerator. This process is at constant pressure and returns the cycle to its starting point.

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*Taconis, K. W. "Techniques of Very Low Temperatures," Leiden, June (1950); also British Patent 635,948 (1950).

In practice the so-called isothermal processes are not truly isothermal and irreversibilities are introduced. However, the modifications introduced in practice do not affect the basic princple of operation. The piston movements are ideally displacements only, involving no work. The valves are both at the high temperature, resulting in reliable operation. The system has the advantage of relative mechanical simplicity and lends itself to cascaded operation.

5.2 PRACTICAL MACHINES

Operational refrigerators based on a modified Taconis cycle using single stage, and cascaded stages compounded with a Linde cycle are produced by Arthur D. Little, Inc.* for refrigeration at various levels down to about 4°K with heat ejection at ambient (300°K). They are referred to as the Gifford-McMahon system.

5.3 COEFFICIENTS OF PERFORMANCE, POWER REQUIREMENTS FOR MACHINES SUITABLE FOR REFRIGERATION OR LIQUEFACTION AT TEMPERATURES OF 77°K AND ABOVE

The coefficient of performance of Taconis cycle refrigerators as exemplified in the Gifford-McMahon system has been presented theoretically by Hrycak.** If p_1 is the high pressure at which gas leaves the cold end of the regenerator and enters the lower (cold) expansion volume, and if p_2 is the low pressure at which the gas exits finally from this volume, then since the piston movement does not work (being a displacer only), the heat absorbed at the cold end of the refrigerator is just the enthalpy change, ΔH, of the gas entering and leaving. If the cold end is isothermal, then ΔH is given by:

$$\Delta H = (p_1 - p_2)\, V \tag{31}$$

where V is the total maximum volume at the cold end.

It is found moreover (see references cited) that the exit gas at the top end is at a temperature, T_E, somewhat above the ambient or sink temperature, T_S, due to the isenthalpic nature of the cycle. Bearing this in mind, the quantity of gas per cycle

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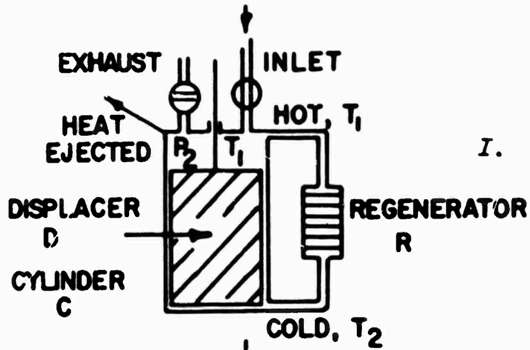
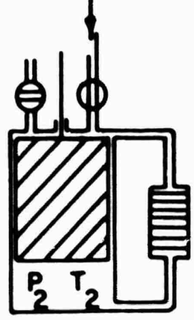
*See Gifford, W. E. and Hoffman, T. E. Adv. in Cryogenic Engineering 6, 82 (1961); Gifford, W. E. and McMahon, H. O. Proc. 10th Intern. Cong. of Refrig. (Copenhagen) 1, 100 (1959) and 1, 105 (1959); McMahon, H. O. and Gifford, W. E. Adv. in Cryogenic Eng. 5, 354 and 368 (1959) and McMahon, H. O. Cryogenics 1, 65 (1960).

**Hrycak, P. Cryogenics 3, 23 (March 1963).



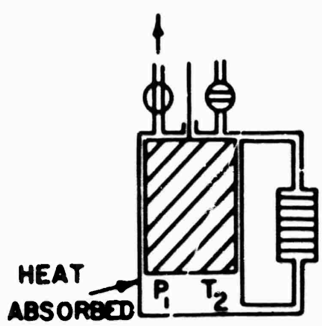
I. *ISOTHERMAL COMPRESSION*

with displacer down, gas is compressed from p_1 to p_2. Heat ejected at hot end of cylinder.



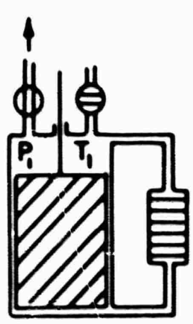
II. *CONSTANT PRESSURE GAS TRANSFER*

Displacers raised to top. Gas transferred to cold end via regenerator. Make up gas from compressor maintains pressure at p_2.



III. *ISOTHERMAL EXPANSION*

with displacer up, exhaust valve open, so that pressure drops to p_1. Heat absorbed at cold end of cylinder.



IV. *CONSTANT PRESSURE GAS TRANSFER*

Displacer lowered to bottom. Gas transferred to hot end via regenerator. Return cycle to starting point.

Fig. 47 - Steps in the operation of an idealized refrigerator using a displacement expander. (The Taconis engine.)

entering (or leaving) the cold end is:

$$M = \frac{p_2 V}{RT} - \frac{p_2 V}{RT_E} \tag{32}$$

where T is the refrigerated temperature of the cold end. If one assumes isothermal compression of the gas exterior to the refrigerator at a temperature, T_S, and assumes that on compression the gas can be assumed to be perfect, then the ideal coefficient of performance, C.P., is:

$$\text{C.P. (ideal)} = \frac{L}{P} = \frac{T}{p_1 T_E - p_2 T} \; \frac{p_1 - p_2}{\ln (p_1/p_2)} \; \frac{T_E}{T_s} \tag{33}$$

where L is the refrigerated load and P the input power. The value of T_E can be determined from knowledge of the other parameters cited and the nature of the working gas.

Evaluations of the C.P. (ideal) have been made for various values of T, assuming helium to be the working gas and assuming, as has been successfully adopted in practice*, that $p_1 = 300$ psig, $p_2 = 75$ psig and $T_s = 300°K$. The data are presented in Table 20.

TABLE 20

VALUES OF C.P. AND η_{rel} FOR GIFFORD-MCMAHON SYSTEM OPERATING DOWN TO 77°K

(For sources, see text)

T	C.P. (Carnot)	C.P. (Ideal)//	Low Power Level		Higher Power Level	
			C.P. (actual)	η_{rel}	C.P. (actual)	η_{rel}
80°K	0.363	0.164	0.059	0.16	0.072	0.198
100°K	0.5	0.206	0.075	0.15	0.091	0.182
150°K	1.0	0.326	0.122	0.12	0.148	0.148
200°K	2.0	0.460	0.167	0.084	0.207	0.104

//Ideal values for Gifford-McMahon system with parameters as stated in text.

*See for example, McMahon, H.O. Cryogenics 1, 65 (1960).

To compute the actual C. P. values which would be encountered in practice and which must be known in order to make fair comparisons with other cycles mentioned earlier in this report, the same assumptions will be made regarding isothermal compressor inefficiency and motor drive inefficiency. It will be assumed (see Section VI.1) that for helium gas the isothermal compressor efficiency is 67% and the motor drive efficiency 80% for the low power levels and 67% and 85% respectively for the higher power levels (~1 kW, as assumed for previous cycles). The losses due to regenerator inefficiencies, heat losses, etc., will be lumped together and a refrigeration efficiency of 70% is assumed for the low power levels and 80% for the higher power levels. These values are taken from measured values published by McMahon et al (loc. cit.). We then have evaluated C.P. (actual), which is tabulated in Table 20 and shown in Fig. 48, as a function of the refrigeration temperature, T. Also in this figure are presented the corresponding values of the figure of merit, η_{rel} (η_{rel} = C.P. (actual)/C.P. (Carnot)).

Estimates of the power requirement, P, for refrigeration at the three temperatures of interest, namely: (1) 77°K (nitrogen reliquefaction), (2) 85°K (fluorine reliquefaction), and (3) 90°K (oxygen reliquefaction) were made from the data of Table 20. The results are shown in Fig. 49, which plots P versus the refrigerative load, L at temperatures of 77°K, 85°K and 90°K, assuming the sink temperature, T_S, to be 300°K. Also in Fig. 49 are shown the Carnot power requirements under the same conditions and the study limits for the refrigerative loads.

It is to be noted that the Gifford-McMahon system, even if operated in an ideal manner with no losses or inefficiencies, does not have an ideal coefficient of performance, C. P. (ideal), equal to the Carnot C.P. (see Table 20). Only in the extreme limit of p_1 tending to equal p_2 does the expression given above for C.P. (ideal) approach asymptotically to the Carnot value. However, in this limit the useful refrigeration (equal to (p_1 - p_2)V per cycle) also tends to zero. As was pointed out by Hrycak (loc. cit.), the McMahon-Gifford cycle is able to increase its thermodynamic efficiency only by decreasing its 'volumetric efficiency', which is the refrigerating effect per unit volume of the system. This is common handicap of nearly all gas refrigerating systems. The notable exception to this rule is the Stirling cycle, which in reversible operation has a Carnot efficiency and this fact renders it particularly suitable therefore for minimum power requirements.

In employing a Taconis cycle engine for liquefaction, the gas to be liquefied would be introduced at approximately atmospheric pressure into thermal contact with the cold end of the refrigerating engine in a separate gas circuit.

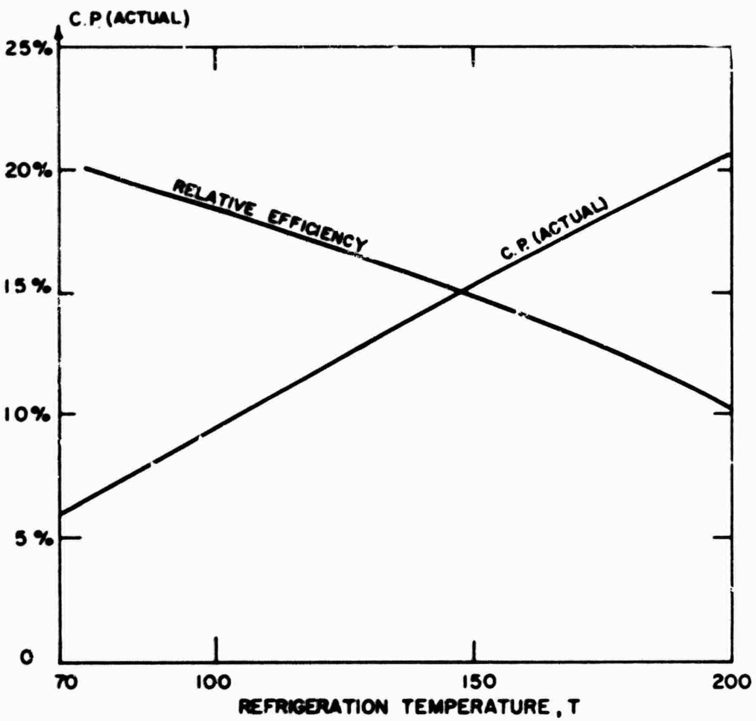


Fig. 48 - Estimates of the coefficient of performance, CP (actual), and the relative efficiency to that of a Carnot engine as a function of the refrigeration temperature, T, for single-stage Gifford-McMahon engines operating down to 70°K. (For source data, see text.) T_R is taken to be 300°K.

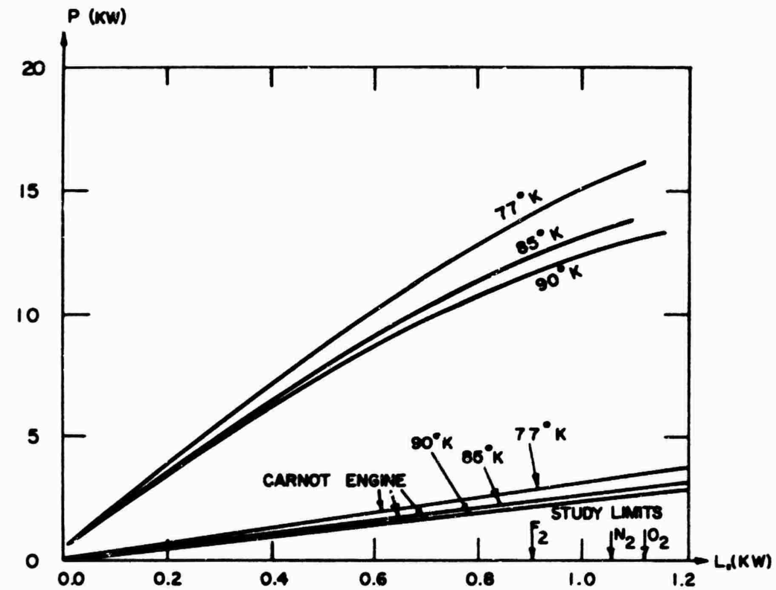


Fig. 49 - Power requirements (kW) versus refrigerative loads, L(kW), for a single-stage Gifford-McMahon engine operating down to 77°K, 85°K and 90°K. T_s is taken to be 300°K. (For sources of data, see text.)

To estimate the power requirements for liquefaction by the Gifford-McMahon system of N_2, F_2 and O_2, it must be observed that the sensible heat of the gas from T_s to its boiling point at about 1 atmos. pressure acts as a full load on the refrigerator in addition to the latent heat load. The data for the computation are given in Table 21 below.

TABLE 21

POWER REQUIREMENTS FOR LIQUEFACTION OF N_2, O_2 AND F_2 USING GIFFORD-MCMAHON SYSTEMS

(For sources, see text)

Substance	ΔH 300°K to B.P. joules/g	ℓ (Latent heat) joules/f	Low Power Level		Higher Power Level	
			C.P. (actual)	P for 100 lbs/day kW	C.P. (actual)	P for 1000 lb/day kW
N_2	236	199	0.056	4.08	0.068	33.5
F_2	173	172	0.063	2.87	0.076	23.8
O_2	192	213	0.067	3.18	0.081	26.3

From the above data the curves of Fig. 50 were drawn which show the power P required for liquefaction as a function of the liquefaction rate in liters per hour for N_2, F_2 and O_2.

The liquefaction figures of merit in terms of kW-hr per lb of substance liquefied are given in Table 22.

TABLE 22

LIQUEFACTION FIGURE OF MERIT FOR GIFFORD-MCMAHON SYSTEMS OPERATING DOWN TO 77°K

(For sources, see text)

Substance	Boiling Point	Liquefaction Figure of Merit	
		(lower power levels)	(Higher power levels)
N_2	77°K	0.98 kW-hr/lb	0.805 kW-hr/lb
F_2	85°K	0.69 kW-hr/lb	0.57 kW-hr/lb
O_2	90°K	0.765 kW-hr/lb	0.63 kW-hr/lb

All the above data are conservative ratings of the power requirements based on current practice. It is considered that in the period 1965-70, by reducing inefficiencies of these machines, the power requirements could be reduced by about 15% below those stated.

5.4 CASCADED MULTIPLE GIFFORD-MCMAHON ENGINES USED IN COMPOUND SYSTEM

As with the Stirling cycle, the Gifford-McMahon engine lends itself readily to cascading for going to very low temperatures. Not many examples of such systems are currently available and in consequence the method is best illustrated by the three-stage cascaded Gifford-McMahon engine combined with a closed loop J-T He circuit for refrigeration down to liquid helium temperatures as produced by A. D. Little Inc. (See the brochure "Cryodyne Maser Refrigerator" issued by A. D. Little Inc., which brochure also describes the system. Other descriptions also have been provided elsewhere*). It consists of three expander volumes operating at 80°K, 35°K and 15°K, connected together with regenerators and all operating from the same compressed He gas supply in phase and in the manner described above for the single stage engine. The inlet and exhaust pressures to the expanders are 280 and 80 psia respectively and the total gas flow for a system (Model 1, as described by McMahon in Cryogenics 1. 65. 1960) providing 4 watts of refrigeration at 4.2°K is 15 scfm. Each expansion stage of the engine serves as a precooling step for a closed-cycle J-T He circuit which has inlet and exhaust pressures of 280 and 15 psia and a flow of 2 scfm for the 4 watt refrigerator. The closed-loop J-T circuit has heat exchangers between the ingoing and outgoing He streams between all temperature levels.

Considerable detail on the performance of this system has been provided in the reference cited and a theoretical study of the thermodynamic behavior has been made by Hrycak.** If one assumes that for Model I, providing 4 watts of refrigeration at 4.2°K, the full 10 hp of the compressor motor is required, then the minimum coefficient of performance is 0.0054. This clearly is a conservative figure. Other sizes of helium temperature refrigerators based on the same compound cycle are marketed by A. D. Little.

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*See, for example: Gifford, W. E. and Hoffman, T. E. Adv. Cryo. Eng. 6.82. 1961. McMahon, H. O. Cryogenics. 1. 65. 1960.
**Hrycak. P. Cryogenics, 3. 23. 1963.

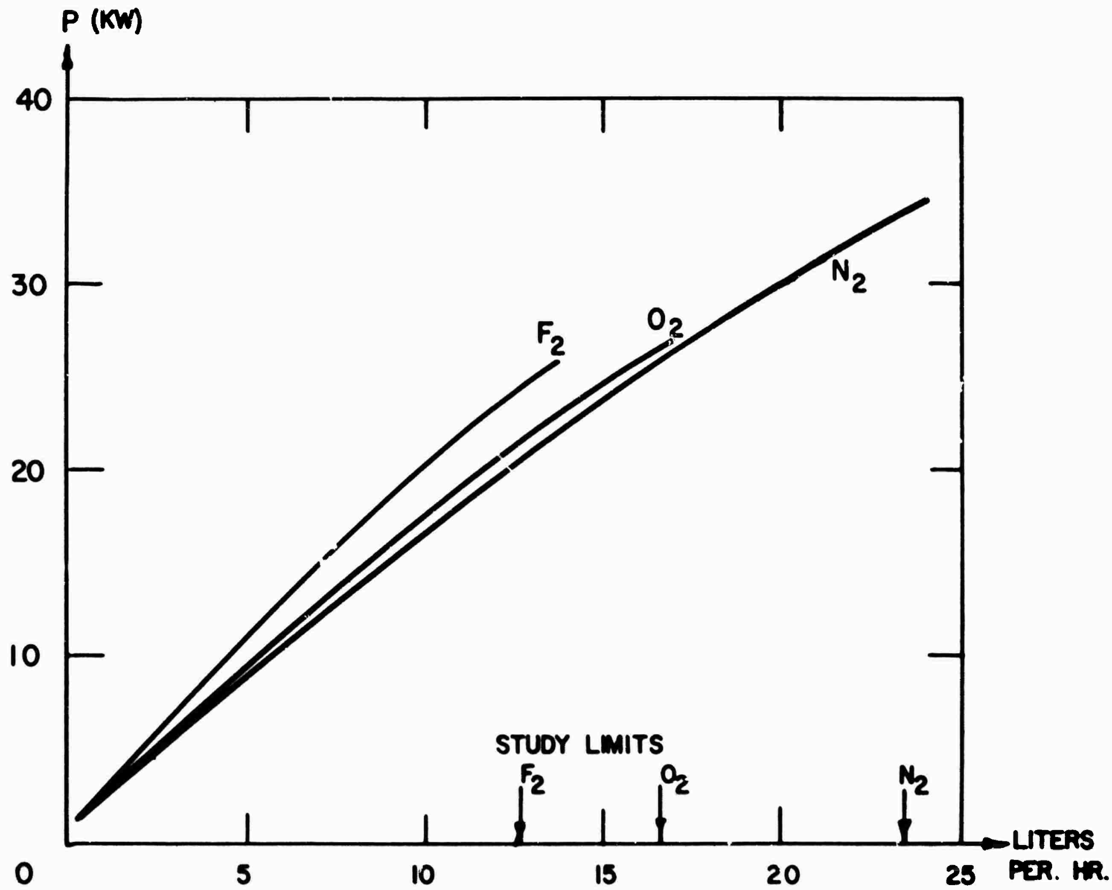

Fig. 50 - Power requirements (kW) for liquefaction of N_2, F_2 and O_2, using single-stage Gifford-McMahon engine versus rate of liquefaction (liters/hr.). T_s is taken to be 300°K. (For source of data, see text.)

It is clearly possible to design other arrangements of compound cycles using Gifford-McMahon engines to provide, for example, refrigeration at H_2 temperatures, liquefaction of H_2 and/or He. They would follow the same general scheme as that outlined above for the helium temperature refrigerator. There is insufficient data available to the authors at present, however, to allow presentation of their possible performances. However one may note that they would possess the same advantageous features as the single stage Gifford-McMahon engines, as outlined above.

V.6 MISCELLANEOUS SYSTEMS

6.1 ROEBUCK DEVICE

In 1945 J. R. Roebuck* proposed a new type of refrigerator, which is described briefly by Scott.** A diagram illustrating the principle of this device is shown in Fig. 51. The working fluid, a gas, flows in the crank-shaped pipe from A to B to C to D, as indicated by the arrows on the diagram. The pipe is rotated rapidly. The motion of the crank causes compression in leg B, the heat of compression being removed by a cooling medium. Expansion occurs isentropically in leg C, which is insulated. An external compressor or blower is required to complete the cycle from D to A.

To the authors' best knowledge, a working model has not as yet been developed. To be successful very high peripheral speeds would be required which imposes severe mechanical design problems.

6.2 VORTEX TUBE.

The Vortex Tube*** is a device seemingly quite unrelated to other schemes of refrigeration. A simple illustration of the device is shown in Fig. 52. A gas at a pressure of the order of magnitude of 5 or 6 atmospheres enters the tube tangentially through a nozzle as shown by the arrows and expands, rotating and attaining a high velocity. The gas may escape to the left, through the unobstructed full diameter, or through the small centrally disposed aperture to the right. The ratio of the amounts of gas which take each path is determined by the adjustment of a throttling valve at the end of the left hand line. It is observed that the gas which emerges through the small aperture is much colder than the gas entering the tube, while the gas which emerges to the left is somewhat warmer. It has been possible to separate air at 20°C and a few atmospheres pressure into the cold and warm streams of -50°C and 200°C, respectively.****

It would be possible to make a refrigerating machine incorporating the Vortex Tube as the source of refrigeration, in a manner analogous to using an expansion engine or a valve.

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*Roebuck, J. R. J. Appl. Phys. 16, 285 (1945).
**Scott, R. B. Cryogenic Engineering, publ. Van Nostrand Co. (1959), p. 38.
***Rangue, G. J. U. S. Patent No. 1952281, December 6, 1952.
****Hilsch, R. Rev. Sci. Instr. 18, 18 (1947). English Translation by Estermann, I.

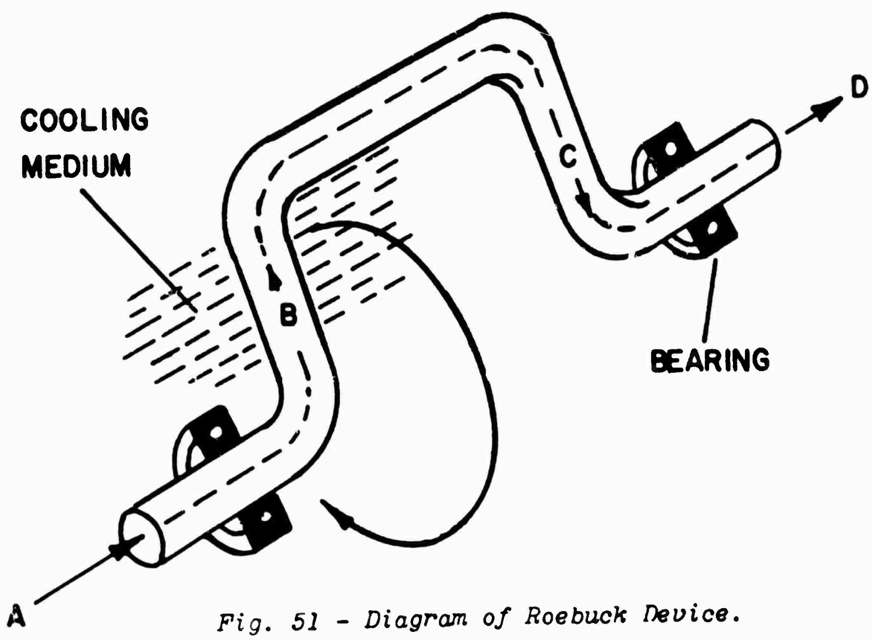

Fig. 51 - Diagram of Roebuck Device.

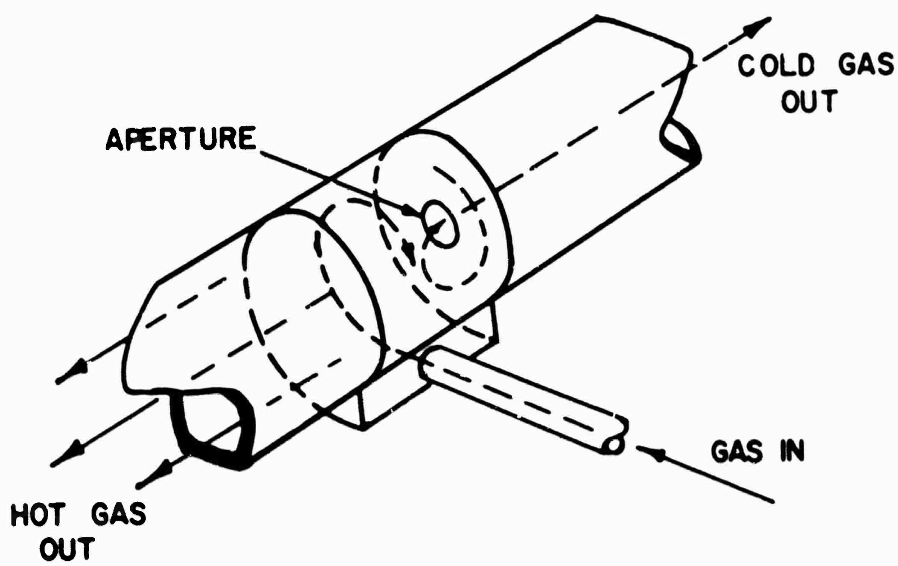

Fig. 52 - Diagram of vortex tube.

The performance of such a refrigerating machine operating on this cycle has been calculated,* using data from Hilsch. Further, the performance for cooling to about -50°C has been compared to an analogous machine using an adiabatic expansion engine and it was found that the Vortex Tube performance suffers by the comparison, being at least 13 times less efficient. It may be concluded that the Vortex Tube, while having the advantage of simplicity, is not efficient enough to be of potential wide-spread use as a refrigerating device.

6.3 DESORPTION METHODS

The desorption method of cooling and/or liquefaction for obtaining very low temperatures was proposed by Simon** and first put into practice for helium liquefaction by Mendelssohn*** in 1931. It has not, however, come into general use since that time due to the superiority of other methods.

The method is dependent on the use of materials such as charcoal, silica gel, "molecular sieve" materials, etc., which are well known to absorb large quantities of gas per g. of material at low temperatures.****

A brief description of the method used by Mendelssohn for the liquefaction of helium follows: Activated charcoal was placed in a container, surrounded by a vacuum jacket, which container had a pipe leading from it vertically upwards so that it could be connected either to a supply of low pressure helium gas or to a vacuum pump. The system was placed in a bath of liquid hydrogen and a small quantity of "exchange gas" was introduced into the vacuum jacket to permit thermal contact between the charcoal container and the liquid hydrogen. In this condition, helium gas was introduced into the container where it was absorbed by the charcoal up to a pressure of about one atmosphere. The heat of absorption was taken up by the liquid hydrogen. The vacuum jacket was then evacuated of exchange gas, so adiabatically isolating the charcoal container, and the charcoal container was pumped by a fast vacuum pump. This pumping desorbed the helium gas from the charcoal and the process resulted in a cooling of the absorbent to approximately 4°K. This cold container then served to permit condensation of liquid helium in a separate container thermally attached to the charcoal container.

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*Daunt, J. G. Handbuch der Physik, XIV, 1 (1956).
**Simon, F. Phys. Zeit. 27, 790 (1926).
***Mendelssohn, K. Zeit. f. Phys. 73, 482 (1931).
****See, for example, Itterbeek, A. V. and Dirgenen, W. V. J. Phys. Radium 11, 25 (1940), for data for helium on charcoal.

The adsorption method of cooling, as described above, is a "one-shot" process, which places it in a poorly competitive position for most refrigerative applications with a continuous cooling method. To date, no continuous closed cycle desorption refrigeration process has been proposed or developed and it is considered that this adsorption technique is unlikely to become of immediate practical significance for continuous liquefaction or reliquefaction of gases.

6.4 SOLID STATE DEVICES

6.41 THERMOELECTRIC COOLING

McCormick* has studied the potentialities of thermoelectric cooling in the range below 77°K. It is clear that existing knowledge does not permit the construction of practical thermoelectric coolers in the temperature region of liquefied gases. However, it is conceivable that some future materials breakthrough could lead to thermoelectric devices being used for gas liquefaction.

The basic phenomenon on which thermoelectric coolers are based is the following: Whenever a circuit composed of two dissimilar conductors is so situated that the junctions are at different temperatures, an electromotive force can be observed in the circuit. This, of course, is the familiar effect upon which thermocouples are based. It has as its corollary the fact that an emf applied across the junctions, if they are thermally isolated, will cause them to be at different temperatures, the current through the circuit causing heat to be evolved at one junction and absorbed at the other. Thus, a schematic diagram for a thermoelectric refrigerator would be as shown in Fig. 53.

The effectiveness of a thermoelectric device is a function of the properties of the thermoelements. These properties are the Seebeck voltage, the electrical resistance and the thermal conductance. There has been postulated** a dimensionless term, called the figure of merit. The expression for the figure of merit, M, is given by:

$$M = \frac{T}{4} \frac{\alpha_{PN}^2}{R_{PN} K_{PN}} \tag{34}$$

*McCormick, J.E., "Cryogenic Thermoelectric Cooling," ASTIA Document #AD259770, May 1961.

**For a fuller account, see, for example:

Ioffe, A. F. "Semiconductor Thermoelements and Thermoelectric Cooling," Publ. Infosearch Ltd., 1957.

Goldsmid, H. J. "Applications of Thermoelectricity," Publ. Methuen and Co., 1960.

MacDonald, D. K. C. "Thermoelectricity," Publ. Wiley and Sons, 1962.

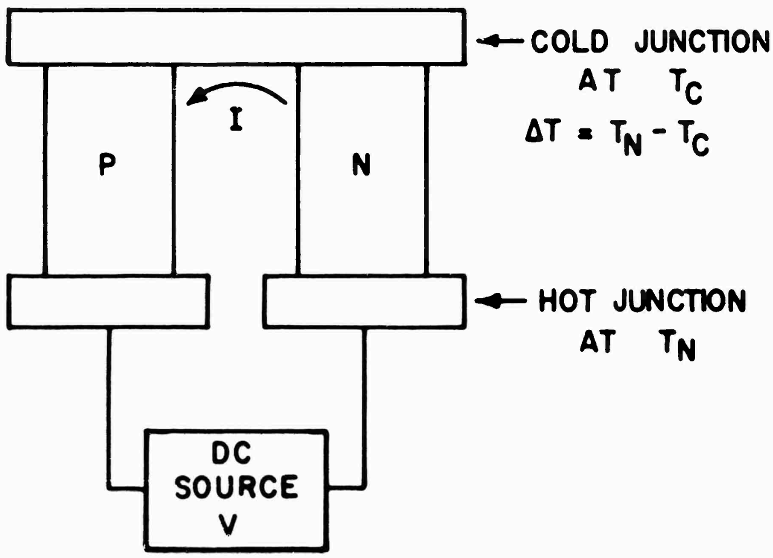

Fig. 53 - Schematic diagram for a thermoelectric refrigerator.

where T = temperature, absolute,

α_{PN} = Seebeck voltage for couple,

R_{PN} = Electrical resistance of circuit,

K_{PN} = Thermal conductance of circuit.

In terms of the properties of individual thermoelements (assuming joint resistances to be negligible),

$$R_{PN} = \frac{\rho_P \, l_P}{A_P} + \frac{\rho_N \, l_N}{A_N} \qquad (35)$$

where p = electrical resistivity of each element,

l = length of element,

A = cross-sectional area of element,

and

$$K_{PN} = \frac{K_P A_P}{l_P} + \frac{K_N A_N}{l_N} \qquad (36)$$

where K = thermal conductivity of each element.

If the two elements are arranged in the optimum configuration, then it can be shown that

$$\frac{\alpha_{PN}^2}{R_{PN} \, K_{PN}} = \frac{(\alpha_p - \alpha_N)^2}{(\sqrt{\rho_P K_P} + \sqrt{\rho_N K_N})^2} \qquad (37)$$

It is convenient to relate this expression to Ioffe's definition of his figure of merit, Z, defined such that

$$Z = \frac{4M}{T} = \frac{(\alpha_P - \alpha_N)^2}{(\sqrt{\rho_P K_P} + \sqrt{\rho_N K_N})^2} \qquad (38)$$

It has been demonstrated that, for any couple, the maximum temperature difference between the heat sink and the object being cooled is given by the expression:

$$\Delta T_{max} = 2M_1 \, T_{sink} \qquad (39)$$

where M_1 is the effective figure of merit and is defined by

$$M_1 = \frac{Z}{4} \frac{2T_{sink} - \Delta T}{2} \qquad (40)$$

For materials tested at 77°K the best values of Z are in the range 0.5 x 10^{-3} to 5 x 10^{-3} (i.e., M_1 from 0.01 to 0.10). Recently Smith and Wolfe* have shown that Bi-Sb alloys can reach the higher figure given here. Even so, the use of such materials with a single junction device (single stage) yield only small temperature drops. Fig. 54 shows the maximum (for optimized configuration) drop in temperature, ΔT_{max}, for a single junction produced by a single stage thermoelectric refrigerator plotted against the mean temperature, T_M. The lower curve in the figure is for Z = 1.5 x 10^{-3}, as found for many commercial products. The middle curve is for Z = 5 x 10^{-3} and the upper curve represents what might be expected in possible future development. If one assumes an M_1 value of 0.05, then 16 stages would be required to go from 77°K to 20°K and 31 stages to go from 77°K to 4°K. Such a device seems too complex to be practical.

At ambient conditions, a value of M_1 = 0.3 (i.e., Z = 4 x 10^{-3}) is fairly common. If we assume that we can achieve this figure of merit at lower temperatures, then it would be possible to go from 77°K to 20°K in four stages. It has been derived that the coefficient of performance for each stage is given by the expression:

$$CP_{max} = \frac{\sqrt{1 + 4M_1} - 1 - \frac{\Delta T}{Tc}}{\sqrt{1 + 4M_1} + 1} \tag{41}$$

Using this expression for CP and the formula $\Delta T = 2M_1 T_{sink}$, it may be calculated that it requires ten thousand watts of power to deliver one watt of refrigeration at 20°K with a heat sink at 77°K.

It may be concluded that at the present state of knowledge of the properties of thermoelements, thermoelectric cooling is not competitive with existing gas systems for supplying refrigeration in the temperature region of liquefied gases from the points of view of efficiency or practicality.

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*Smith, G. E. and Wolfe, R. Bull. Amer. Phys. Soc. 2, 137 (1961).

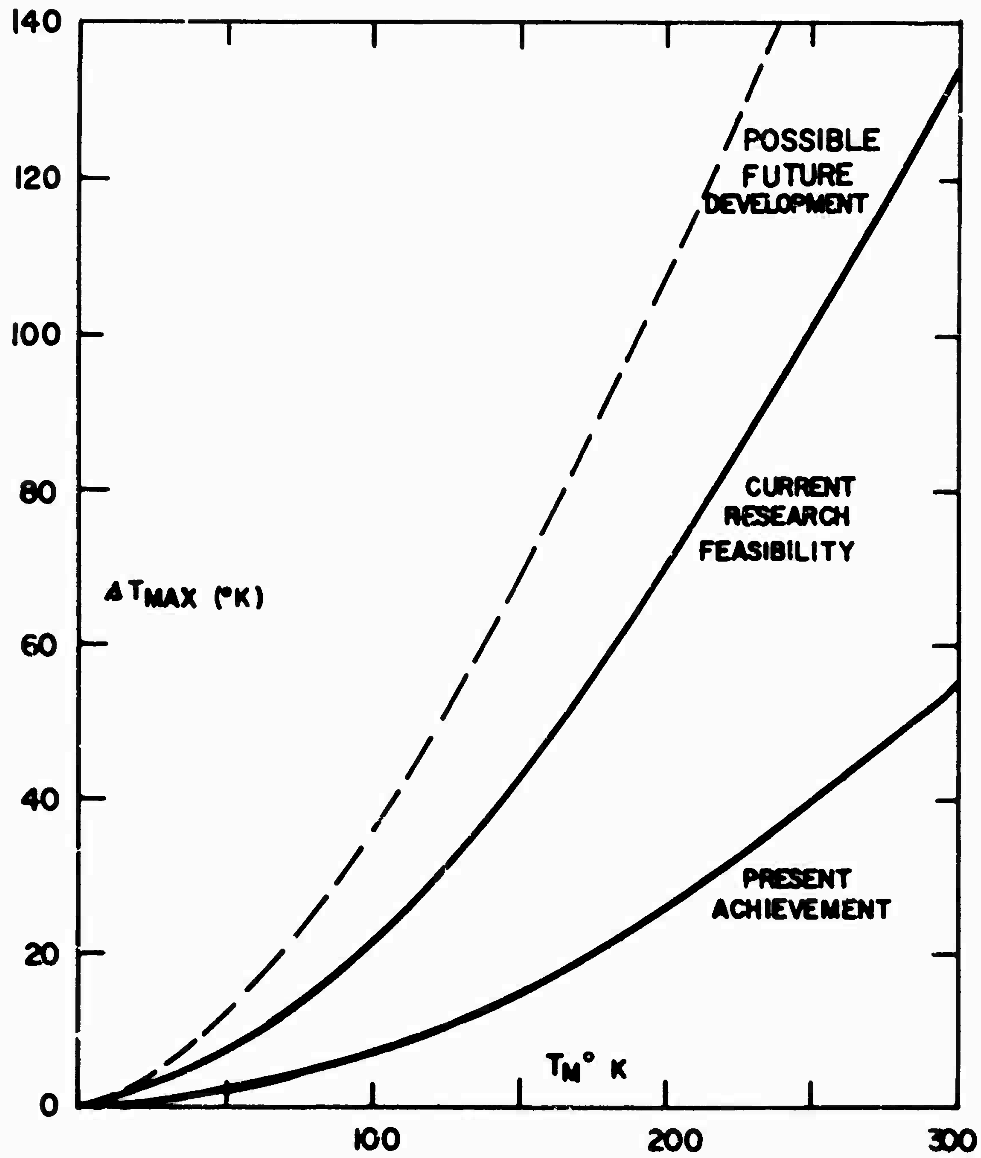

Fig. 54 - Maximum temperature drops for single-stage thermoelectric coolers.

6.42 GALVANO-THERMOMAGNETIC REFRIGERATORS

The theory of galvano-thermomagnetic refrigerators has been discussed by many authors, notably O'Brien and Wallace, by Harman and Honig, by El-Saden, by Norwood and by Delves.* The theories cover many possible arrangements of systems and the three of the simpler arrangements will be discussed here. Following Harman and Honig, consider the two element device for longitudinal mode of operation illustrated in Fig. 55. It consists of two rectangular paralellopiped bars designated as ℓ and r respectively. The right hand bar is a p-type semiconductor and the left hand an n-type. The ends of the bars are thermally attached to two reservoirs. The upper reservoir at the temperature T_1 constitutes the "cold-head"; that is to say, the refrigerated load. The lower reservoir, at the temperature T_2, is the heat sink, such that $T_2 > T_1$. The ends of the semiconducting bars at the lower temperature end (T_1) are connected electrically for operation in the longitudinal mode discussed in this sub-section. A magnetic field, of intensity H, is externally applied at right angles to the current flow, along the z axis of Fig. 55. An electric current is fed into and out of the lower ends of the bars, which are both maintained at temperature T_2 but which are electrically insulated from each other.

The theory of such a refrigerator as presented by Harman and Honig** results in a calculated optimal coefficient of performance, CP_L, given by:

$$CP_L = \frac{T_1}{(T_2 - T_1)} \cdot \frac{(\Lambda_L - T_2/T_1)}{(1 + \Lambda_L)} \tag{42}$$

where Λ is given by the expression:

$$\Lambda_L^2 = 1 + \frac{1}{2} (T_1 + T_2) (P_\ell - P_r)^2 / R_L K \tag{43}$$

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*O'Brien, B. J. and Wallace, C. S. J. Appl. Phys. 29, 1010 (1958).
Harman, T. C. and Honig, J. M. J. Appl. Phys. 33, 3188, 3178 (1962). and ibid. 34, 189 (1963). See also Harman, T. C. J. Appl. Phys. 34, 239 (1963) and Appl. Phys. Lett. 2, 13 (1963).
El-Saden, M. R. J. Appl. Phys. 33, 1800 (1962).
Norwood, M. H. J. Appl. Phys. 34, 594 (1963).
Delves, R. T. Brit. J. Appl. Phys. 13, 440 (1962).
**Harman, T. C. and Honig, J. M. J. Appl. Phys. 33, 3188 (1962). See also same authors, J. Appl. Phys. 33, 3178 (1962) and 34, 189 (1963).

where P_ℓ and P_r are the Ettinghausen-Nernst coefficients* for the left hand and right hand bars, R_L the total electrical resistance of both bars in series and K is the rate of heat flow (in the absence of an electric current through the device) along the two bars in parallel per unit temperature difference across their ends. The device can be operated "adiabatically" or "isothermally," meaning in the first case the bars are adiabatically isolated along their lengths and in the second case they are in thermal contact so that $\partial T/\partial y = 0$. For each mode of operation the appropriate adiabatic or isothermal Ettinghausen-Nernst coefficients must be used in equation (43).

As in the discussion of thermoelectric cooling, it is convenient to introduce a "figure of merit." Harman and Honig give for the optimum figure of merit, Z_L^{opt}, for this longitudinal mode of operation:

$$Z_L^{opt} = \frac{(P_\ell - P_r)^2}{\left[(k_\ell \rho_\ell)^{1/2} + (k_r \rho_r)^{1/2}\right]^2} \qquad (44)$$

where k_ℓ and k_r are the average thermal conductivities of the left and right hand bars and ρ_ℓ and ρ_r are the average electrical resistivities of the left and right hand bars.

It is to be noted that in equation (42) for the coefficient of performance the first term, $T_1/(T_2 - T_1)$, on the right hand side is the ideal Carnot refrigerator coefficient of performance.

An arrangement for a two element device for operation in the transverse mode is shown schematically in Fig. 56. Again it consists of two rectangular parallelopiped bars designated and r respectively. The ends of the bars are thermally attached to two reservoirs. The upper reservoir at the temperature T_1 constitutes the "cold-head"; that is to say, the refrigerated load. Note, as is shown in Fig. 56, that the bars are electrically insulated from each other at this end, as they are also at the end at temperature T_2. The center points of both bars closest to each other are electrically connected by a shorting bar, and electric current is fed into the center point on the outside of bar ℓ and carried away from the center point on the outer side of bar r. A magnetic field of intensity H is externally applied at right angles to the current flow, along the z axis of Fig. 56.

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*For definitions of the various galvano-magnetic coefficients, see: Mazur, P. and Prigogine, I. J. Phys. Radium 12, 616 (1951).

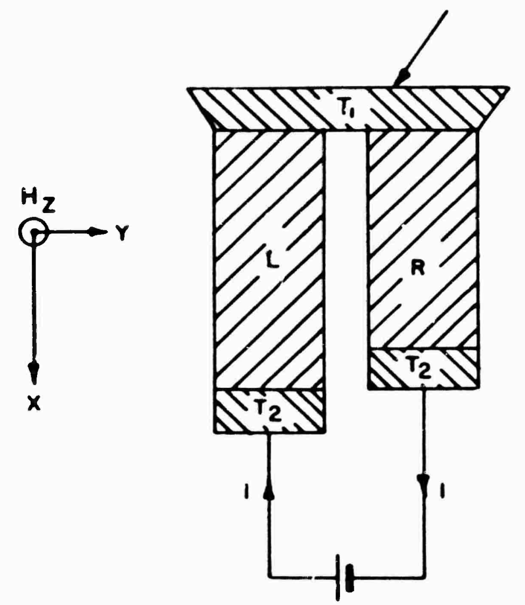

Fig. 55 - Schematic of galvano-thermomagnetic refrigerator operating in the longitudinal mode.

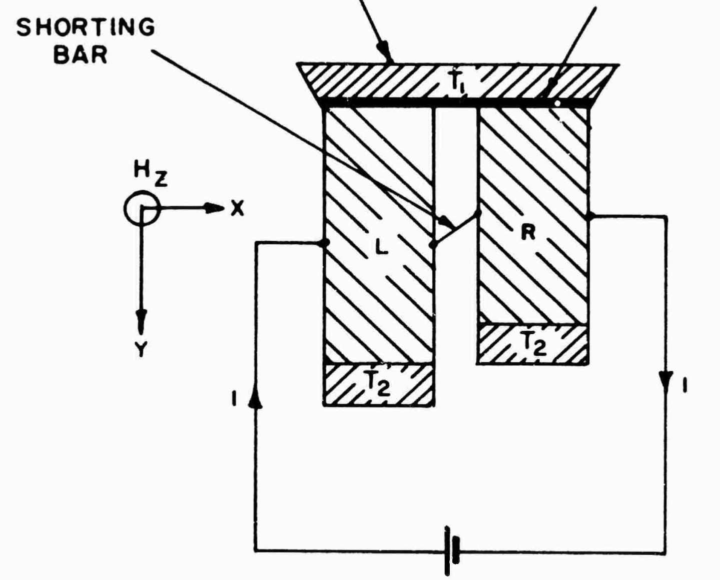

Fig. 56 - Schematic of galvano—thermomagnetic refrigerator operating in the transverse mode.

As found by Harman and Honig, the optimum coefficient of performance, CP_T, in this transverse mode of operation is given by:

$$CP_T = \frac{T_1}{(T_2 - T_1)} \frac{(1 - \Lambda_L T_1/T_2)}{(1 + \Lambda_T)} \tag{45}$$

where Λ_T is given by:

$$\Lambda_T = 1 - \frac{1}{2} T_2 H^2 N^2 / R_T K \tag{46}$$

where N is the average transverse Nernst coefficient, R_T the total electrical resistance of both bars in series in the y direction and K is as defined earlier.

The optimum figure of merit for this mode of operation is:

$$Z_T^{opt} = H^2 N^2 / K_\rho \tag{47}$$

where the symbols have their previously defined significances.

The maximum temperature differences which can be established are given by:

$$\text{Longitudinal:} \quad \Delta T_{max} = \frac{(P_\ell^2 - P_r)^2 T_1{}^2}{2KR_L} = \frac{1}{2} Z_L T_1{}^2 \tag{48}$$

$$\text{Transverse:} \quad \Delta T_{max} = \frac{H^2 N^2 T_2^2}{2KR_T} = \frac{1}{2} Z_T T_2{}^2 \tag{49}$$

Where Z_L and Z_T are the longitudinal and transverse figures of merit.

These are to be compared with the expressions given earlier for the maximum temperature difference attainable by thermoelectric cooling, which for small temperature differences yield the approximate expression:

$$\text{Thermo-electric:} \quad \Delta T_{max} \approx \frac{(\alpha_P - \alpha_N)^2}{\left[(\rho k)_P^{1/2} - (\rho k)_N^{1/2}\right]^2} = \frac{1}{2} Z\overline{T^2} \tag{50}$$

A much simpler configuration for a galvano-thermomagnetic refrigerator is the one discussed for example by O'Brien and Wallace and by Delves.* The simple device is a single element of an intrinsic semiconductor operating in the transverse mode, as schematically illustrated in Fig. 57. In this the electric current is along the x axis, the applied magnetic field along the y axis and the consequent heat flow is along the z axis. This arrangement has the advantages of great simplicity and of the requirement for one working material only. The maximum temperature difference obtainable is as for the two-element transverse systems given by $\Delta T_{max} = (1/2)\, Z_T T^2$ where $Z_T = H^2 N^2/k\rho$.

Unfortunately the highest figures of merit observed so far for either the transverse or longitudinal modes of operation are small. For example Wolfe and Smith** recently reported Z for the longitudinal case equal to $7.3 \times 10^{-3}/°K$ for a Bi-Sb alloy at 100°K in a magnetic field of 700 gauss. Cuff et al***, also using Bi-Sb alloys, report similar figures and have experimentally obtained therewith temperature drops of about 5°K, starting at 77°K. For other data, see previous references cited. Kooi et al# report ΔT values of 35°K for transverse operation in the range between 200°K and 100°K also using Bi-Sb. However for a transverse mode refrigerator to cool from 80°K to 4°K, it would require $Z_T \approx 10/T$ to achieve a coefficient of performance of one tenth that of a Carnot refrigerator, which values of Z_T are far beyond any reasonable extrapolation of current developments in galvano-thermomagnetic materials. These results seem to indicate that galvano-thermomagnetic refrigerating devices, although theoretically they should have figures of merit of about ten times better# than thermoelectric cooling devices, are still not competitive with gas cooling systems and the comments made earlier concerning such thermoelectric devices are applicable to galvano-thermomagnetic refrigerators operating in longitudinal and transverse modes. Whereas by the development of better semiconductor materials with higher figures of merit, it may be possible in the future to use galvano-thermomagnetic devices for cryogenic refrigeration, at present they clearly require multiple cascading (or shaping##) to achieve any significant temperature drops, and this results in inefficient devices. Moreover these galvano-thermomagnetic devices all require their accompanying magnetic fields which may use further power if permanent magnets are not used. (the use of superconducting magnets in persistent modes of

*See references given earlier.

**Wolfe, R. and Smith, G. E. Bull. Amer. Phys. Soc. 7. 173. (1962).

***Cuff, K. F. Horst, R. B. Weaver, J. L. Hawkins, S. R. Kooi, C. F. and Enslow, G. M. Appl. Phys. Letters. 2.145. (1963).

#Kooi, C. F. et al. Lockheed Missiles and Space Co. Tech. Report No. ASD-TDR-62-1100. Mar. 1963.

##For discussion on "shaping" see O'Brian and Wallace. (loc. cit.) and Kooi, C. F. et al. in above reference.

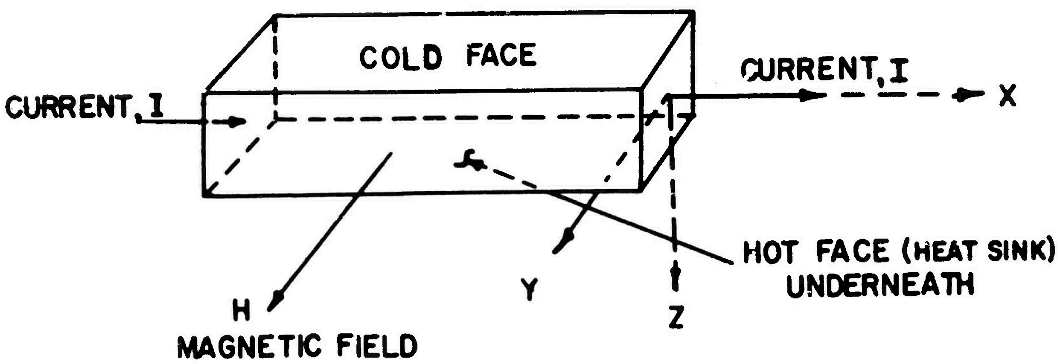

Fig. 57 - Schematic arrangement of single element transverse galvano-thermomagnetic refrigerating device.

operation would obviate additional electrical power requirements, but they in turn need low temperatures for their operation, the maintenance of which is power consuming.)

6.5 MAGNETIC COOLING

This process of the cooling of paramagnetic salts by adiabatic demagnetization* is used for obtaining temperatures well below 1°K, starting from temperatures obtainable with liquid helium (1°K to 4°K). Although it is not applicable for refrigeration** or liquefaction of the subject gases in this report, a brief review of the technique is included here, largely in order to point out the reasons for the inapplicability of the magnetic cooling method at temperatures well above 4°K.

A paramagnetic salt, for example chromium potassium alum, is used as the working substance. It is located in a space (vacuum jacket) which can either be highly evacuated or contain "exchange" gas and which is cooled to as low a temperature, T_1, as possible by a bath of liquid helium ($T_1 \approx 1°K$). The steps in the process can be followed by use of the entropy-temperature diagram of Fig. 58. Initially the vacuum jacket contains some exchange gas so that the working paramagnetic salt is maintained isothermally at the bath temperature, given by the point a on the T-S diagram. The salt is then magnetized isothermally along the path a-b on the diagram, where, for example, b lies on an isofield line of, say, 10,000 gauss. At this point the vacuum jacket is pumped to high vacuum, thus isolating thermally the working salt from its surroundings. The external magnetic field, H, is then reduced to zero and the salt follows the isentropic path b→c, so reducing its temperature to T_f.

Typical values of T_f obtainable with chromium potassium alum for various initial conditions of T_1 and H_b are shown in Table 23.

The success of the method is dependent on the fact that the entropy change on magnetization, ΔS_{mag}, is large compared with the total entropy, S_1, the system at T_1 and $H = 0$. For example, for $T_1 = 1°K$ and $H_b = 20KG$, $\Delta S_{mag}/S_1$ is 83% for iron ammonium alum and 78% for chromium potassium alum and in

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*For general descriptions see, for example: de Klerk, D. Handbuch der Physik, Vol. XV, p. 38 (1956): Casimir, H. B. G. "Magnetism and Very Low Temperatures," Cambridge Phys. Tracts (1940); Garrett, C. G. B. "Magnetic Cooling," Harvard Univ. Press (1954).

**For continuous magnetic refrigeration below 1°K see: Daunt, J. G. Proc. Phys. Soc. (London) B 70, 641 (1957) and Heer, C. V. Barnes, C. B. and Daunt, J. G. Rev. Sci. Instr. 25, 1088 (1954).

..nsequence very low temperatures, corresponding to the low entropies on magnetization, can be achieved. This result that $\Delta S_{mag}/S_i$ can be so high is due to the fact that for the paramagnetic salts used in the magnetic cooling method the contribution to the total entropy due to lattice vibrations is negligible compared to the magnetic contributions when T_i is of the order of 1°K. At higher temperatures the ($T > 10$°K) lattice entropy and hence the total entropy, S_i, becomes much larger and additionally ΔS_{mag} becomes much smaller. ($\Delta S_{mag} = F(H_b/T_i)$). These two effects result in very small values of $\Delta S_{mag}/S_i$ and hence yield very small temperatures changes on adiabatic demagnetization.

TABLE 23

FINAL TEMPERATURES OBTAINABLE BY MAGNETIC COOLING OF CHROMIUM POTASSIUM ALUM

T_i(°K)	H_b(KG)	T_f(°K)
1.12	10	0.08
1.13	15	0.06
1.11	20	0.033

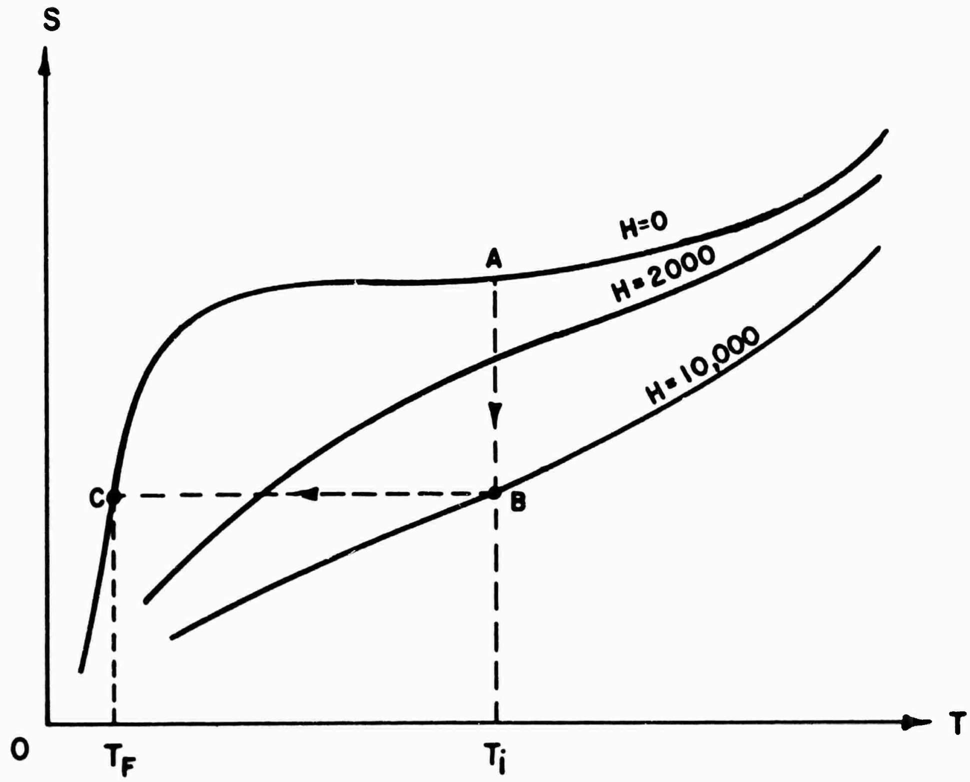

Fig. 58 - Schematic of steps taken in magnetic cooling system plotted on an S-T diagram.

VI. SOME DATA ON COMPRESSORS, MOTORS AND EXPANDERS

VI.1 COMPRESSORS

In evaluation of compressors, we are concerned with size and weight, efficiency, reliability and adaptability to space environment. Reliability and adaptability to space environment are functions of the type of compressor employed, while for each type of compressor size and weight and efficiency are functions of throughput and compression ratio.

There are two major classifications of compressors: reciprocating and centrifugal. In addition, other designs such as the bellows compressor have been proposed to meet special problems, particularly those of miniaturization. In general, centrifugal machines are more efficient than reciprocating in large throughputs with low compression ratios. Reciprocating compressors are more adaptable to high compression ratios and also to miniaturization.

1.1 RECIPROCATING COMPRESSORS

Data have been obtained from various manufacturers on the performance characteristics and sizes of currently available reciprocating compressors. A partial list of manufacturers includes Cooper-Bessemer Corp., Ingersoll-Rand, Sulzer Bros., Inc., Air Products and Chemicals, Inc., York Corp., Worthington Corp., Norwalk Co., Westinghouse Air Brake Co., Gardner-Denver Co., Pressure Products Industries and Chicago Pneumatic Tool Co. For some models, performance data are not available for the compressor separately, but only for the unit as a whole.

A common assumption in the literature, when making studies of comparative cycles or other studies in which compressor efficiency is a factor, is to assume that the isothermal efficiency of compression is some constant amount, such as typically 70%. The isothermal efficiency is defined as the ratio of the work required in isothermal compression from the input condition to the work actually required.

An interesting and useful format for expressing compressor performance is shown for example in Fig. 59 (Taken from data supplied by Ingersoll-Rand Co., sheet 9500.10-2. 1960). Here the power per unit rate of volume flow is plotted versus the compression ratio with the specific heat ratio, as a parameter. Inlet pressure is assumed to be 14.4 psia.

For a perfect gas, the work of isothermal compression is given by the equation:

$$W = -PV \ln \frac{P_2}{P_1} \qquad (51)$$

If we assume that we are compressing a perfect gas, we may then take points on Fig. 59 and test the assumption of constant isothermal efficiency over the range of operation. This test leads to the result that the isothermal efficiency is roughly constant for a constant value of γ. Results of this calculation are shown in Table 24.

TABLE 24

ISOTHERMAL EFFICIENCY VARIATION OF INGERSOLL-RAND RECIPROCATING COMPRESSORS

(Taken from data distributed by Ingersoll-Rand Co. For reference, see text).

Ratio of Specific Heats	Isothermal Efficiency	
	Compr. Ratio = 2	Compr. Ratio = 5
1.10	74	78
1.2	72	74
1.3	70	69
1.4	68	67
1.7	66	60

This analysis leads to the expectation that the power requirements of all reciprocating compressors will follow roughly the same pattern. The isothermal efficiency will be mainly a function of the gas properties and, for thermodynamic calculations on cycles, may be considered nearly independent of compression ratio. It is seen that, of the fluids of interest in this study, relatively lower isothermal efficiencies may be anticipated with helium, and monatomic gases in general, than with the other fluids. It is seen in the section on expanders also that reciprocating expanders are less efficient when used with monatomic than with diatomic gases.

The minimum size of the machines covered by Fig. 59 is about one horsepower (745 watts). It should be noted that the power requirements indicated are brake horsepower, from the motor to the compressor shaft. In computation of the total input power requirements the motor efficiency and the motor-compressor drive efficiency must be taken into account also. Motor efficiency is defined as the percentage of power supplied to the motor which

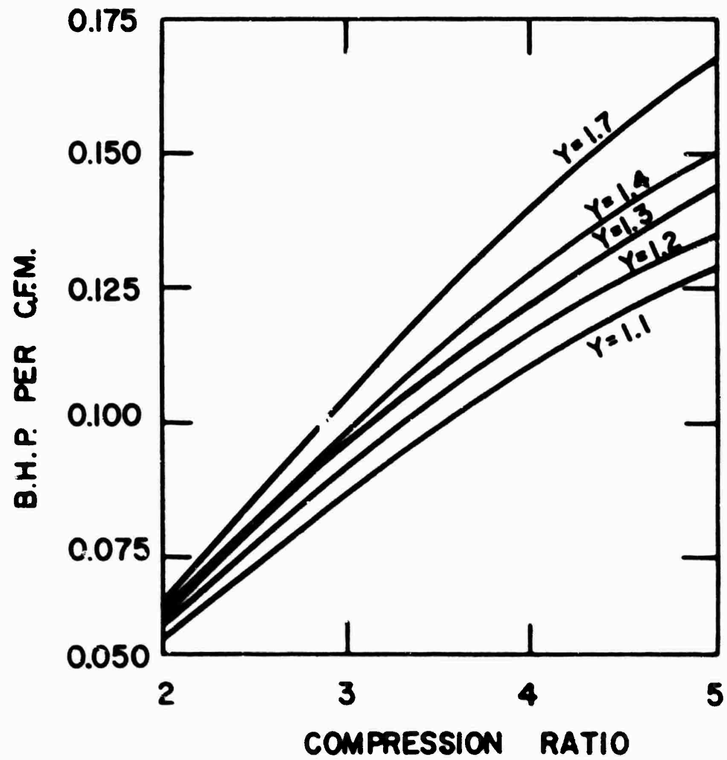

Fig. 59 - Power requirements for reciprocating compressors. (For source, see text.)

actually reaches the shaft. Estimates of this can be made for three phase, 60 cycle motors from Table 25 collated from data provided by Reliance Motors.

Weights of reciprocating compressors are a function of both horsepower and pressure ratio. Data on sizes and weights of two and three-stage compressors including motor, in the size range of 20 to 150 horsepower, are presented in Tables 26 and 27 (source: Ingersoll-Rand Form 3250-C). The curves of Fig. 59 apply to these machines. Data on smaller machines manufactured by Gardner-Denver are shown in Table 28. Perhaps more indicative of the weights of potentially space-borne reciprocating units are the weights of lighter, air-cooled units which are currently marketed as being suitable for intermittent duty only, because of the heat dissipation problem. Performance data on some of these machines manufactured by Ingersoll-Rand are tabulated in Table 29. Dimensions and weights of these machines are tabulated in Table 30 including driver.

TABLE 25

MOTOR EFFICIENCIES

(Data provided by Reliance Motors)
3 Phase, 60 Cycle, 40°C Rise
(Protected - Design B)

HP	Sync Speed RPM	Efficiency 4/4 of full load	Efficiency 3/4 of full load
1	1800	84.5%	81.5%
	1200	77.5	78
	900	76	76
2	3600	81	80.5
	1800	81	81.5
	1200	77	80
	900	77	77
10	3600	80	79
	1800	85	85
	1200	85.5	88
	900	88	89
20	3600	87	87.5
	1800	90.5	91.5
	1200	89.5	91
	900	86.5	87

TABLE 25 (Cont'd)

MOTOR EFFICIENCIES

(Dat r rovided by Reliance Motors)
3 rnase, 60 Cycle, 40°C Rise
(Protected - Design B)

HP	Sync. Speed RPM	Efficiency	
		4/4 of full load	3/4 of full load
30	3600	87.5	88
	1800	92	91.5
40	3600	88.5	90
50	3600	90	92
	1800	90.5	92
	1200	90	91.5
	900	88.5	90
75	3600	-	-
	1800	92.5	93
	1200	90.5	92
100	3600	-	-
	1800	91	92
125	3600	-	-
	1800	92.5	92.5
150	3600	-	-

TABLE 26

SOME DATA ON TWO-STAGE INGERSOLL-RAND COMPRESSORS

(Source: Ingersoll-Rand Co., Form 3250-C)

Piston Displ. cfm	Max. Pres. psig	Motor HP.	Dimensions Including V-Belted Motor feet-inches			Approx. Weight* lbs.
			L	W	H	
130	350	30	8-1	2-10	3-8	2570
226	350	50	9-4	3-0	4-3	3710
386	350	100	15-8	3-7	4-11	7480
504	350	125	17-4	3-10	5-7	8500
94.7	500	30	8-1	2-10	3-8	2370
177.5	500	40	9-4	3-0	4-3	3710
289	500	75	15-8	3-7	4-11	7280
421	500	125	17-4	3-10	5-7	9900

TABLE 27

SOME DATA ON THREE-STAGE INGERSOLL-RAND COMPRESSORS

(Source: Ingersoll-Rand Co., Form 3250-C)

Piston Displ. cfm	Max. Pres. psig	Motor HP	Dimensions Including V-Belted Motor feet-inches			Approx. Weight* lbs
			L	W	H	
86.6	1500	30	11-3	2-10	3-6	2625
165.5	1500	60	13-1	3-4	4-3	4550
255	1500	75	17-7	4-1	4-11	8322
383	1500	150	20-1	4-6	5-6	11,400
64.8	2500	20	11-3	2-10	3-6	2675
124	2500	50	13-1	3-4	4-3	4460
200	2500	75	17-7	4-1	4-11	8490
311	2500	125	20-1	4-6	5-6	11,100

*Boxed for domestic shipping.

TABLE 28

SOME DATA ON GARDNER-DENVER WATER-COOLED COMPRESSORS

(Data from Gardner-Denver Corp.)

Displacement cfm	Pressure	HP*	Size (Inches)** L	W	H	Weight*** lbs
32	100	5	48	20	33	925
64		10	71	20	35	1540
131		25	74	24	41	2455
184		30	75	30	43	3090

*Nominal HP of motor.
**Includes compressor, base, coupling and motor.
***Includes compressor, base, coupling, motor and skids for shipping.

TABLE 29

SOME DATA ON INGERSOLL-RAND HIGH PRESSURE COMPRESSORS

300 TO 1000 PSIG

Symbol	Piston Displacement Cu. ft. per min.	300 psig BHP#	500 psig BHP	1000 psig BHP
231	7.4	1.9	2.3	-
223	6.75 8.6 9.55	- - -	- - -	2.7 3.1 3.9
41	12.7 15.2	3.3 3.9	3.6 4.4	4.1 -
7T2	36.0	9.0	10.2	-
15T2	31.4 41.2 49.5	9.35 - 14.6	10.5 - 16.4	12.6 16.4 -

#BHP = Horsepower to compressor shaft from motor.

TABLE 29 (Cont'd)

SOME DATA ON INGERSOLL-RAND HIGH PRESSURE COMPRESSORS

1500 TO 5000 PSIG

Symbol	Piston Displacement Cu. ft. per min.	1500 psig BHP*	2000 psig BHP	2500 psig BHP	3000 psig BHP
223	6.75 7.6 9.55	2.8 3.25 4.1	2.9 3.4 4.3	3.1 3.55 4.5	3.2 3.7 4.7
15T3	25.0 23.2 17.4	10.7 9.55 7.15	11.3 10.15 7.6	11.9 10.55 7.95	12.3 11.0 8.2
4R15	23.55	-	10.7	11.2	11.6

*BHP = Horsepower to compressor shaft from motor.

TABLE 30

SOME DATA ON INGERSOLL-RAND HIGH PRESSURE COMPRESSORS

DIMENSIONS AND WEIGHTS, INCLUDING MOTOR

Symbol	Dimensions, in.			Weight#, lbs.
	L	W	H	
231	36	20	19	290
41	44	22	25	500
7T2	52	24	31	690
15T2	59	28	34	1150
15T3	58	28	33	1070
4R15	42	30	35	850

#Boxed for domestic shipment.

From the data tabulated above, the weights of reciprocating compressors complete with motors and drives have been estimated as a function of power input (kW) and the results are shown in Fig. 60 in which the shaded area indicates the range in weight. Also in the figure are shown the weights of the motors, for 60 cps, 3 phase motors, which have been taken from data supplied by the General Electric Co. (Bulletin 4-62(15-M)200).

Fig. 61 shows the range of compressor volumes complete with motor and drive as a function of power input (kW) taken from the data tabulated above. Fig. 62 shows the volumes of 60 cps, 3 phase motors as a function of input power (kW) taken from data supplied by the General Electric Co. (Bulletin 4-62(15-M) 200).

All the data presented here, both in tabular and graphical form, are conservative ratings based on currently available compressors and motors. Considerable reductions in weights and volumes, particularly in the volumes, could be anticipated by specialized custom design and by the use of lightweight materials, although it is difficult to estimate exactly what these future optimized figures may be.

A further comment should be made here concerning the fact that all the compressors discussed here use oil lubrication. As is discussed elsewhere in this report, it is desirable that compressors for space systems be oil-free to permit their operation in any attitude, to obviate the weights and volumes associated with filters, etc., which otherwise would be necessary to remove the oil entrained in the working gas and to ensure reliable block-free operation of the cryogenic systems. It is known that developments are in progress on oil-free compressors,* particularly for small sizes, and it is anticipated that this must be further accelerated to meet the requirements of space refrigerators and liquefiers.

1.2 CENTRIFUGAL COMPRESSORS

Centrifugal machines are preferable for large-volume flows accompanied by a low over-all pressure ratio of perhaps 6 or 7. For high pressure ratios, the multiplicity of staging makes centrifugal machines less desirable from a size standpoint. Some data concerning centrifugal air compressors, supplied by Joy Mfg. Co., De Laval Turbines, Inc. and the Worthington Corp., are set out in Table 31 and Figs. 63 and 64 show respectively the weights and volumes of centrifugal air compressors complete with motors as a function of shaft input power (kW). The curves were drawn

*See for example: Ritter, U. et al. J. Refrig. (Mar.-Apr. 1959), p. 37 and references to be found in the "Bibliographic Guide to Refrigeration," 1953-1960 of the Internat. Inst. of Refrig. and Manufacturers brochures.

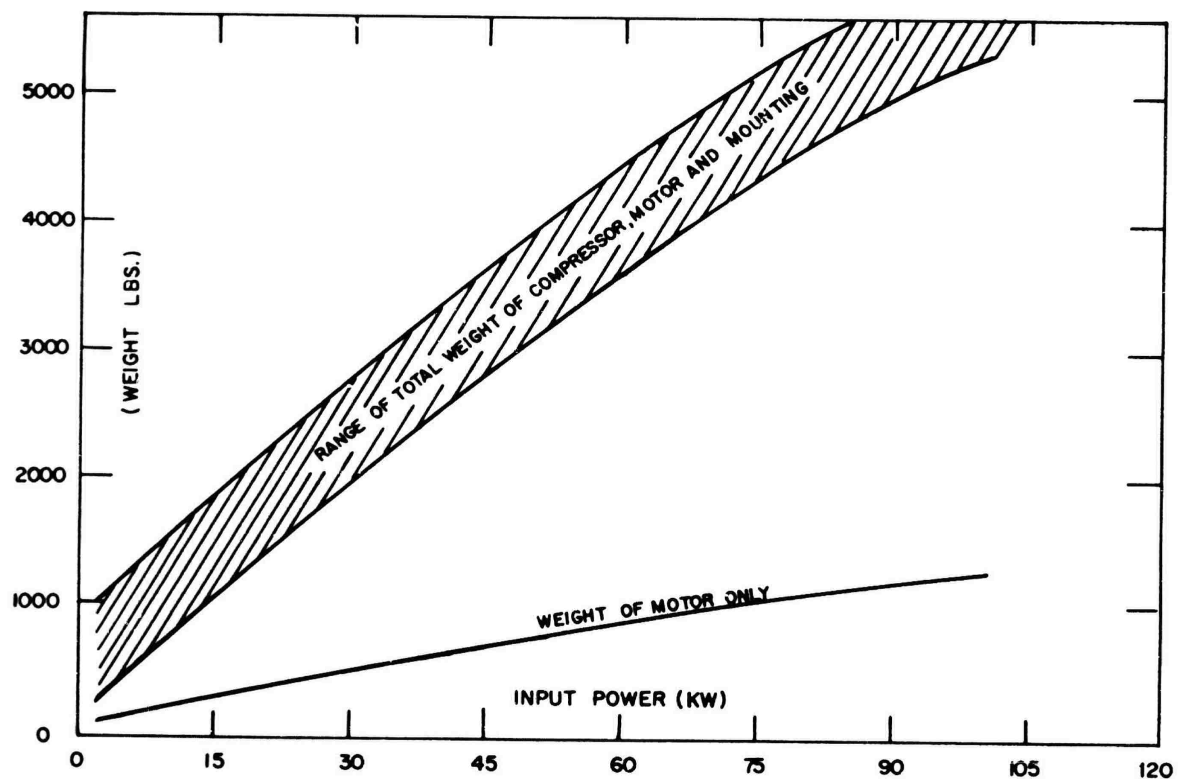

Fig. 60 - Approximate weights of reciprocating compressors, together with their motors and mountings, as a function of the input power to the motor (kW). (For references to sources of data, see text.)

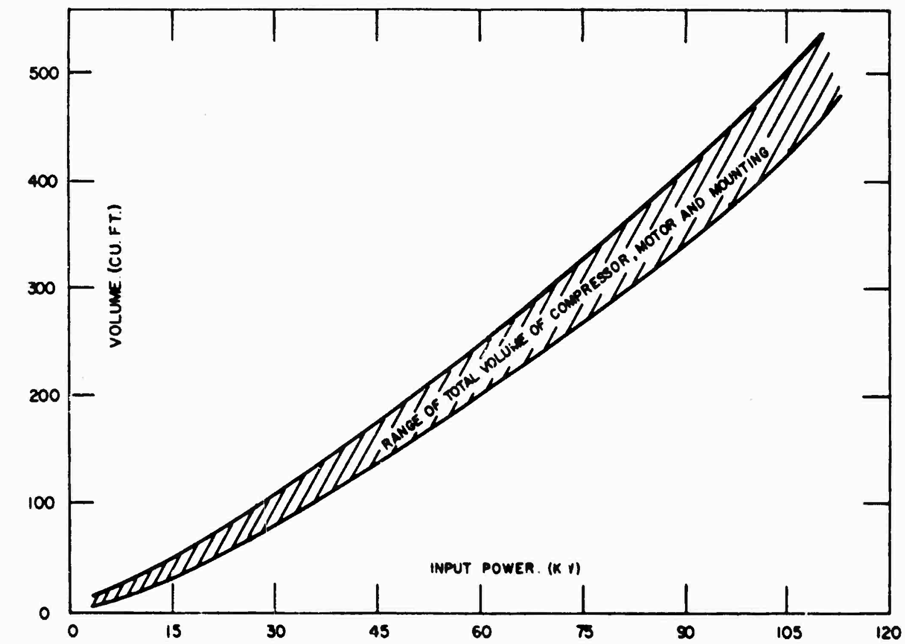

Fig. 61 - Approximate volumes in cu. ft. of reciprocating compressors, together with their motors and mountings, as a function of the input power to the motor (kW). (For references to sources of data, see text.)

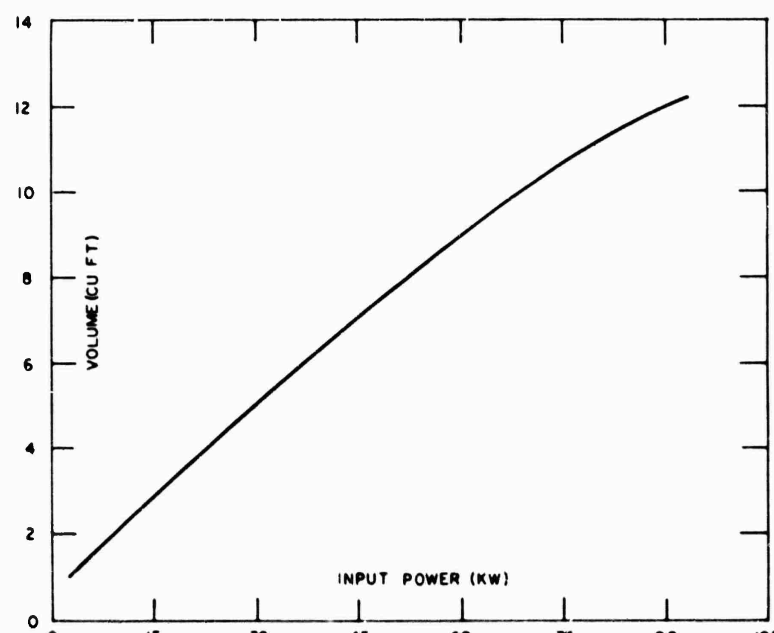

Fig. 62 - Approximate volumes of three phase, 60 cps motors, as a function of their input powers. (For references to sources of data, see text.)

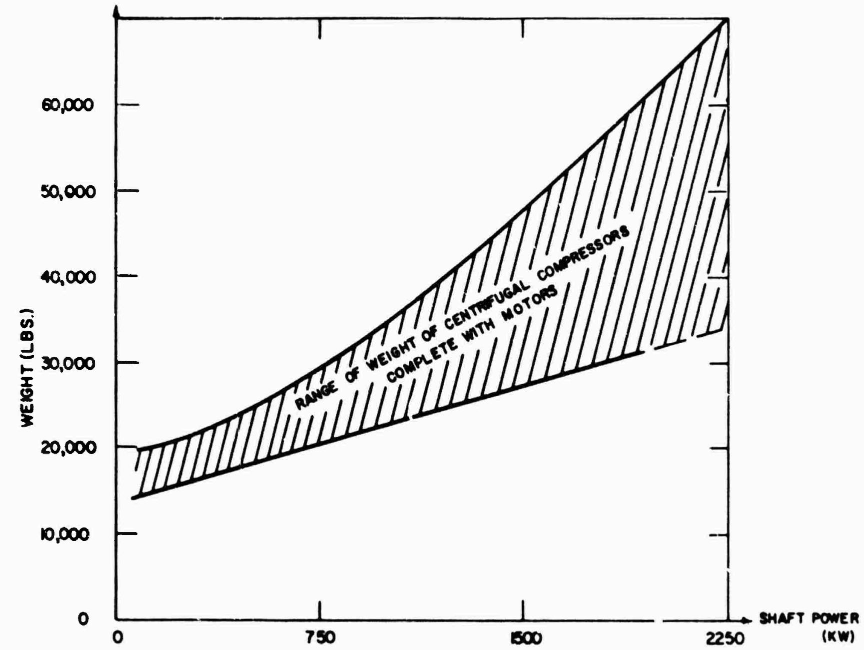

Fig. 63 - Weights of centrifugal air compressors with motors as a function of the input shaft power (kW). (For sources of data, see text and Table VI.1.8)

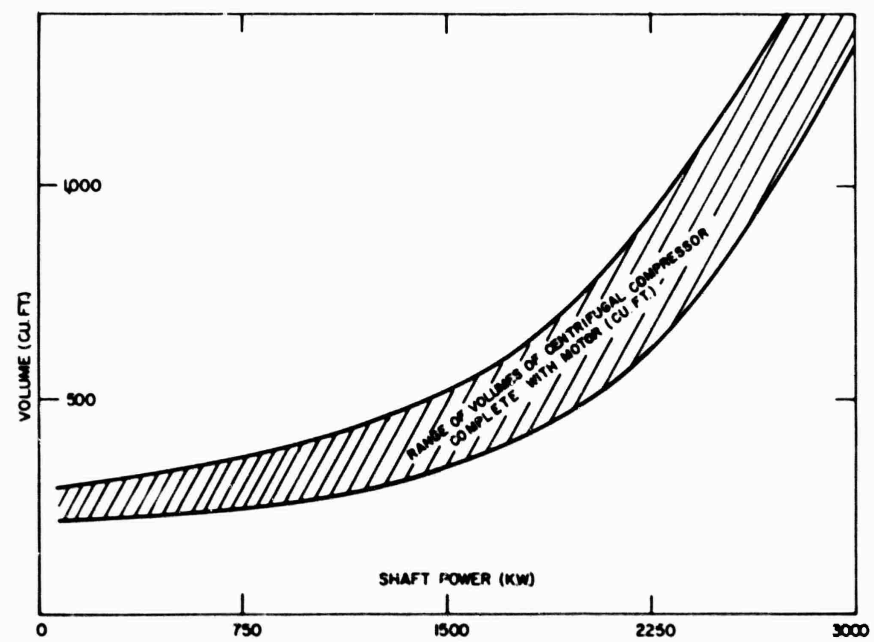

Fig. 64 - Volumes of centrifugal air compressors complete with motors as a function of input shaft power (kW). (For sources of data, see text and Table VI.1.[illegible])

from data quoted in Table 31. The data presented are for currently available items and consequently reflect conservative practice. For space-borne operation considerable savings in weight and volume can be anticipated from future development, due to the use of light-weight materials, etc. It is to be noted that the data presented here are for rather large size compressors, for input shaft powers larger than about 100kW. (Compare Figs. 60 and 61 for reciprocating compressors which give data only up to 100 kW). Developmental effort,* however, is being made to produce centrifugal compressors of smaller sizes and higher pressure ratios because of the ultimate possibility of much higher isothermal efficiency than that obtained with reciprocating compressors. Further, since centrifugal compressors run at higher speeds, motor efficiency will be higher at smaller sizes. A further advantageous feature associated with centrifugal machines is the fact that they can readily be constructed to run with gas bearings, so increasing the reliability of operation.

The smallest centrifugal compressor known to the authors is of approximately three horsepower size and is used in a closed-cycle refrigeration system designed to supply 50 watts of refrigeration at liquid nitrogen temperatures.** It seems possible that in the next few years centrifugal machinery will be applicable to most systems in the range of interest of this study.

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*See for example: "Bibliographic Guide to Refrigeration," 1953-1960, Internat. Inst. Refrig. 1962.

**Hunimoto, W. Y. "Closed-cycle Cryogenic Refrigeration Systems," "Garrett Corporation Paper #AE-2088-R, September 27, 1962.

TABLE 31

SOME DATA ON CENTRIFUGAL TYPE AIR COMPRESSORS

Discharge Pressure psia	Capacity Inlet (A) CFM	Drive Motor BHP (B)	Drive Motor RPM	Length ft. in.	Width ft. in.	Height ft. in.	Weight lb. (E)	Manufacturer and Source of Information
35	500	50	1800	10-0	5-0	5-0	20,000	Joy Manufacturing Co.
95	1000	100	1800	10-0	5-0	5-0	20,000	Joy Manufacturing Co.
135	1000	290	1800	6-0	5-2	4-5	(D) 15,300	De Laval Turbine, Inc.
165	1200	300	1800	10-0	5-0	5-0	20,000	Joy Manufacturing Co.
135	5600	1300	1800	6-11	7-2	(C) 5-8	31,800	Worthington Corp.
135	6200	1400	1800	7-1	7-2	(C) 5-8	34,800	Worthington Corp.
135	7900	1700	1800	7-10	8-2	(C) 7-4	39,600	Worthington Corp.
135	10000	2100	1800	8-6	8-6	(C) 7-4	52,000	Worthington Corp.
265	6000	2400	1800	13-0	7-0	7-0	30,000	Joy Manufacturing Co.
135	12300	2500	1200	9-2	8-10	(C) 7-4	57,400	Worthington Corp.
135	15500	3100	1200	9-10	9-2	(C) 7-4	72,000	Worthington Corp.
135	18400	3700	1200	12-5	11-0	(C) 10-5	85,000	Worthington Corp.
135	23700	4600	1200	13-1	11-9	10-5	104,000	Worthington Corp.

(A) Measured at 14.7 psia and 60°F.
(B) Measured at Motor shaft.
(C) This dimension obtained from comparable De Laval Machine as Worthington literature did not include it.
(D) This figure does not include weight of intercoolers.
(E) Figures are for compressor and accessories but do not include driver.

VI.2 EXPANDERS

A preliminary investigation reveals three basic machines with which we may accomplish expansion. They are the radial inward flow expander, the axial flow impulse expander, and the reciprocating expansion engine.*

The three types of machine all have applications for which they are particularly suited. The radial flow inward expander is good for high volume flow expansion entirely in the gas phase. If it is desired to expand into the liquid region, an axial turbine must be used. This turbine has basically a lower efficiency than the radial one but it is capable of expansion into the liquid region. For low flow rates and/or high pressure ratios the reciprocating expansion engine is the choice. Hansen** cites a break point of 10:1 pressure ratio and 15,000 SCFH flow rate, indicating that below this pressure ratio and above this flow rate expansion turbines are generally used. However, the relatively low wear characteristics and mechanical simplicity of turbines together with the facility they present for use of gas bearings have attracted the attention of designers of small systems. Kunimoto*** mentions an axial turbine for use in a system designed to give 50 watts of refrigeration at liquid nitrogen temperature.

The radial expander is primarily suitable for high flow gas expansion. The lower practical limit of flow is set by structural limitations. In particular, manufacturing limitations set a lower limit on wheel width and an upper limit on turbine speed. Jekat and Nagyszalanczy ("Application and Selection of Industrial Expansion Turbines," Power and Fluids, 1959) set 3/8" as a minimum wheel width. Existing Worthington Radial Expanders are designed to run at 23,500 RPM and have been tested at 40,000 RPM ("Turbine Expanders for Low Temperature Application," Bulletin #G-2933, Worthington Advanced Products Division). Trepp ("Refrigeration Systems for Temperatures below 25°K with Turbo-expanders," Advances in Cryogenic Engineering, Volume 7) claims operating speeds of 92,000 RPM for turbo-expanders in an existing system. Fig. 65 shows minimum throughput versus isentropic enthalpy drop for a radial expander at various speeds. Similar data on existing radial Worthington machines are shown in Fig. 66. At design point, according to Worthington, radial expansion turbines may be expected to have 85% adiabatic efficiency. Hansen indicates a similar number (Fig. 67). The weights and dimensions of the

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*See, for example: Collins, S. C. and Cannaday, R. L. "Expansion Machines for Low Temperature Processes, Oxford Univ. Press, Edmister, W. Chem. Eng. Prog. April 1951, p. 191; Brown, E. H. J. Res. NBS 64c, No. 1, Jan-Mar. 1960, p. 25.
**Hansen, O. A. "Extremely Low Temperatures," Chem. Eng. Feb.23,'59.
***Kunimoto, W. Y. "Closed-cycle Cryogenic Refrigeration Systems," The Garrett Corp., Report No. AE-2088-R.

Worthington expanders of Fig. 66 are shown in Table 32, (Worthington Paper G-2933).

TABLE 32

WEIGHTS AND DIMENSIONS OF SOME WORTHINGTON TURBINE EXPANDERS

(Taken from Worthington Paper G-2933-2)

	Lgth. (in.)	Wdth. (in.)	Hgt. (in.)	Wght. (lbs)
MR-600 - Monobloc design, compressor brake, R-600 turbine	56	47	70	3,500
MR-1200 - Monobloc design, compressor brake, R-1200 turbine	80	91	104	11,000
MA-600 - Monobloc design, compressor brake, A-600 turbine	48	47	64	3,000
CA-600 - Coupled design, compressor brake, A-600 turbine	120	73	65	6,500

For expansion into the liquid region, an axial impulse turbine is required. As a rule of thumb, the lower flow limit for efficient operation of axial expanders seems to be 1/5 to 1/10 of the minimum for radial expanders. At design point an efficiency of 75% for gas expansion in a Worthington axial turbine may be expected (see Bulletin G-2933, Worthington Advanced Products Div.). Hansen indicates a figure of about 79% (see Fig. 67). If liquefaction takes place in the expander, the efficiency is reduced by one-half of the percentage of liquefaction. For example, if the exhaust contains 6% liquid and if dry efficiency is 75%, then the efficiency would be as follows:

$$\left(1.00 - \frac{.06}{2}\right)(.75) = .97\ (.75) = 73\%.$$

In addition, the axial impulse turbine may be turned down to a lower percentage of its capacity flow than the radial reaction turbine. This is because the radial turbine demands a full circle of nozzles. To cut down flow, the flow to each nozzle is throttled, thus causing inefficient expansion in the nozzles. The axial turbine, on the other hand, may be adapted to a wide range of flows by changing the number of nozzles, or admitting flow only to some of the nozzles. Thus, each nozzle operates at design flow rate. The turndown characteristics of

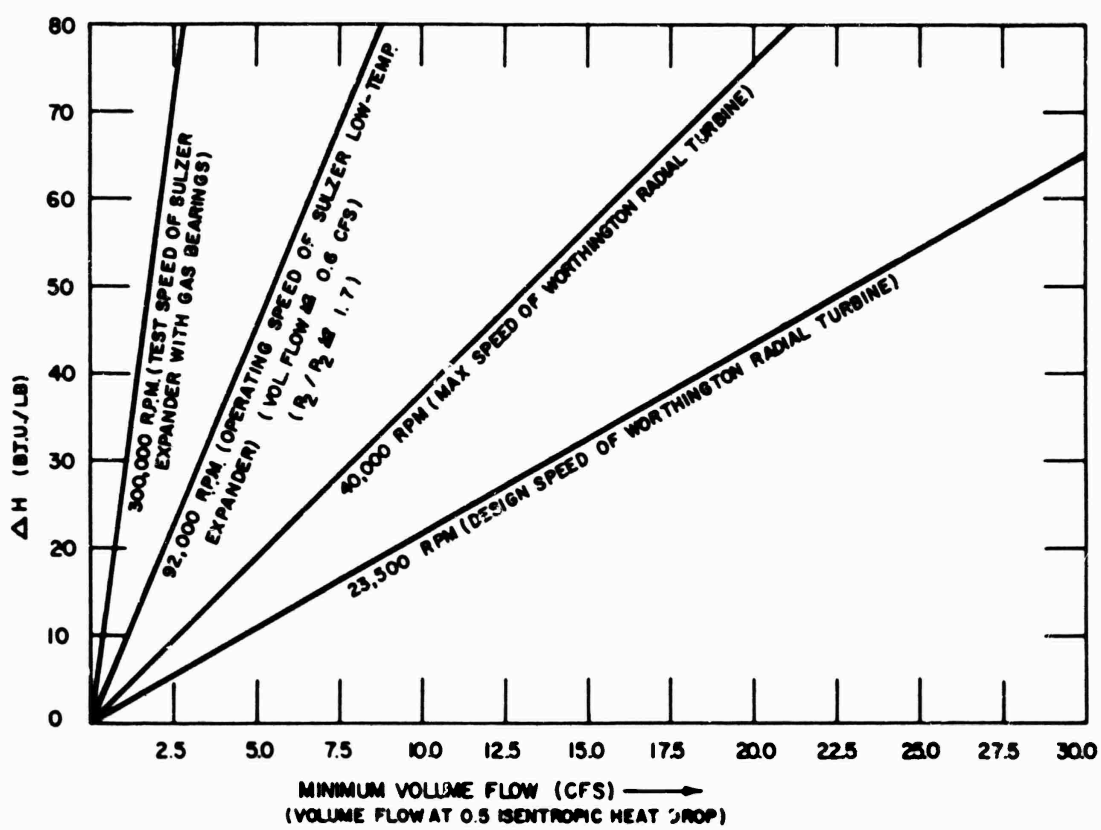

Fig. 65 - Minimum throughput of radial expansion turbine (based on wheel width of 3/8 in.).

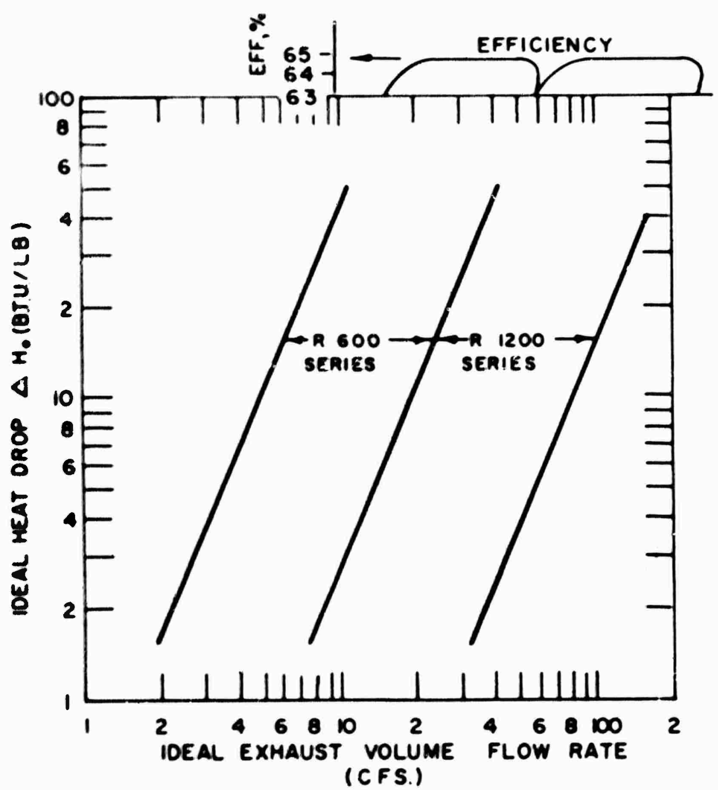

Fig. 66 - Efficiency of Worthington expansion turbines, according to the Worthington Corp., Bulletin No. G-2933.

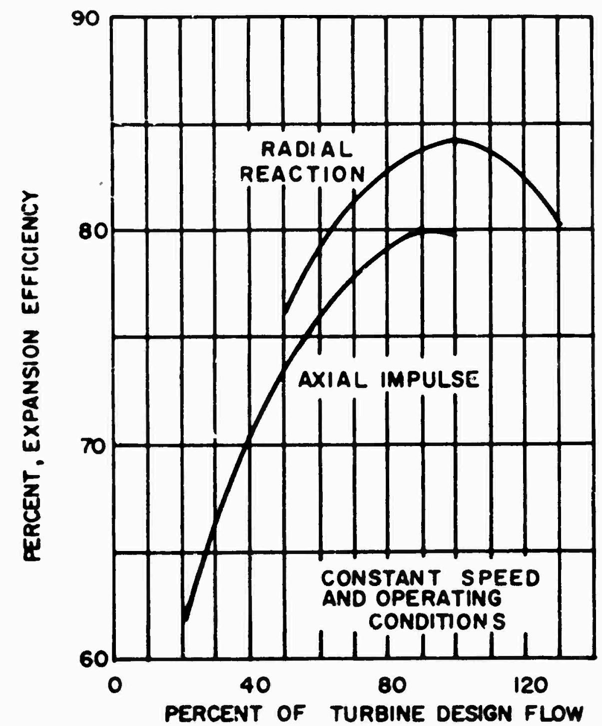

Fig. 67 - Efficiency of radial and axial impulse turbines. (Sources of data, O. A. Hanson, Chem. Eng., Feb. 23, 1959.)

axial and radial turbines are shown in Fig. 68 (from Jekat, "An Impulse Type Expander Turbine," Advances in Cryogenic Engineering, Volume 2).

For flow rates too low for expansion turbines, the flexible rod reciprocating expansion engine is recommended. This type of machine is described by Collins ("Expansion Machines for Low Temperature Processes," Collins and Canaday, Oxford University Press) and in "Infrared Detector Cooling in Ultrahigh Altitude Vehicles," by the Garrett Corp. (ASTIA Report No. 222204). A similar expansion engine is used by Air Products in a small refrigerator ("A Closed-Cycle Helium Refrigerator for 2.5°K", Zeitz and Woolfenden, Cryogenic Engineering Conference, 1962). According to the above references, the efficiencies to be expected are as shown in Fig. 69. Efficiency falls off at low throughputs when the point is reached where further size reduction is impractical and the machine must be run slower to achieve reduced flow. This causes very efficient heat transfer between the gas and the cylinder walls. The lower curves in Fig. 69 are for maximum heat transfer effectiveness. They are approximately the lower limits of efficiency to be anticipated with these machines.

From a standpoint of efficiency, the following machine selections are indicated for expanders:

A. For large, single phase flow the radial reaction should be used. At present, the lowest level of throughput is an exhaust volume of about 10 CFS. This type of machine probably has the potential to be scaled down to 1 CFS. A design point efficiency of 85% may be expected, with turndown characteristics as in Fig. 68.

B. For expansion into the liquid region, the axial impulse turbine should be used. A design point efficiency of 75% may be expected, minus the correction for liquefaction mentioned above. Turndown characteristics are given in Fig. 68. The minimum exhaust volume is about 2 CFS and this type of machine can probably be scaled down by a factor of 10 also.

C. For gas expansion at low flow rates, the flexible rod reciprocating engine is used, with efficiencies as given in Fig. 69. In this case, further increase in RPM and consequently efficiency is blocked by the ability of gas to flow rapidly through very small valve ports, and it seems, therefore, that major improvements in the performance of this type of machine in the smaller sizes are not readily foreseeable.

VII. SPACE ENVIRONMENT AND COMPONENT COMPATIBILITY AND DESIGN

VII.1 The Environment

A partial definition of the space environment* in which the liquefiers or reliquefiers shall be assumed to operate is as follows.

1.1 Altitudes

It will be assumed that the altitudes in which the liquefiers or reliquefiers shall operate range from 100 miles above the earth to 150,000,000 miles (equivalent to Mars or Venus).

1.2 Ambient gas pressure

The ambient gas pressure ranges from that at earth's surface (760 mm Hg) to less than 10^{-9} mm Hg.

1.3 Radiations and plasmas

1.31 Solar wind

This is ionized hydrogen gas with a density $10^2/cm^3$ at 1 A.U. for quiet sun to $10^3/cm^3$ at 1 A.U. for an active sun.

1.32 Stationary gas

The estimate for the density of stationary gas ranges from 10^{-1} to 10^3 protons and electrons per cm^3.

1.33 Solar flares

These are protons of varying energies whose appearance seems to be related to sunspot activity.

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*See: "Space environment for aerospace vehicles." USAF specification bulletin No. 523, November 1960.

"Natural environment of interplanetary space." Shaw, J. W. Ohio State University Research Foundation, WADD Technical Note, January 1960, ASTIA Document No. 250230.

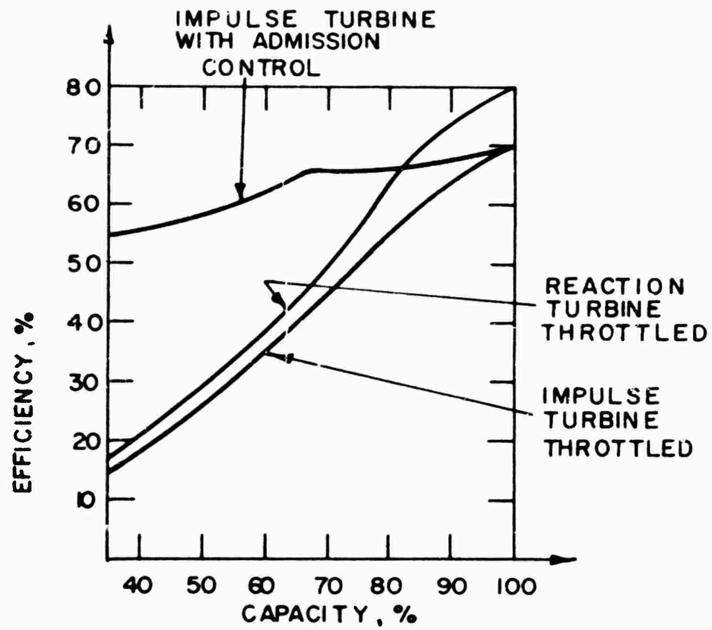

Fig. 68 - Turndown of reaction and impulse turbines, according to Jekat, (Advances in Cryogenic Engineering, Vol. 2, p. 263, 1960).

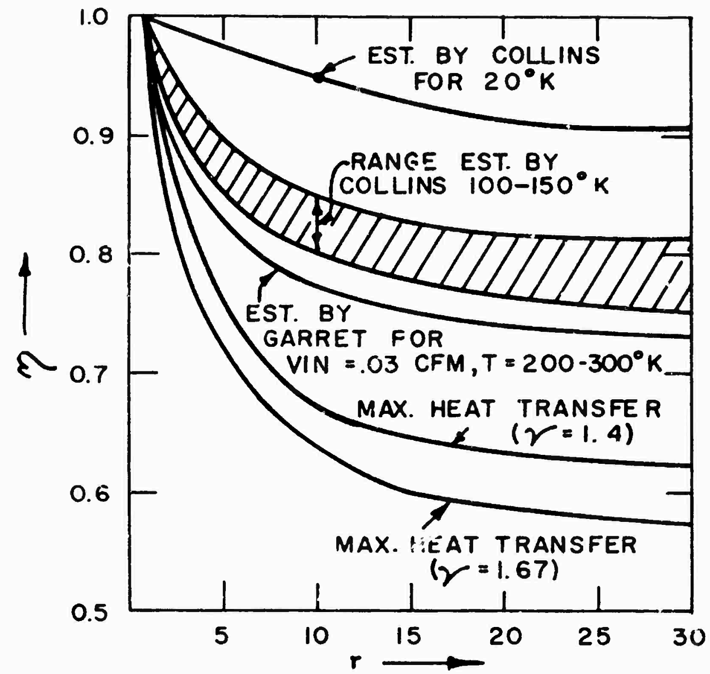

Fig. 69 - Efficiency of flexible-rod expansion engine.

1.34 <u>Cosmic radiation</u>

This consists of 90% protons, 7% alpha particles and the remainder heavier nuclei and electrons. Their energy ranges up to 10^{17} e.v. with an average of approximately 3.6×10^{9} e.v. and their intensity ranges between 1.38 and 2.1 particles/cm^2-sec.

1.35 <u>Van Allen belts</u>

It shall be assumed that the gas liquefiers or reliquefiers considered in this report shall not operate in the Van Allen belts for any extended period of time.

1.36 <u>Nuclear radiation</u>

Such radiation may be present due to possible use of nuclear power in the space vehicle.

1.37 <u>Solar radiation</u>

The solar radiation at a distance 1 A.U. from the sun is about 0.137 watts/cm^2. It is distributed in wavelength as given in Table 33.

TABLE 33

WAVELENGTH DISTRIBUTION OF SOLAR RADIATION

Energy	Wavelength Angstrom Units	% of radiant energy
X-ray and ultra-violet	1 - 2,000	0.2
Ultra-violet	2,000 - 3,800	7.8
Visible	3,800 - 7,000	41.0
Infrared	7,000 - 10,000	22.0
Infrared	10,000 - 20,000	23.0
Infrared	20,000 - 100,000	6.0

1.4 Meteoroids

Some data concerning meteoroids is summed up* as follows.

1.41 Origin

Approximately 90% are of cometary origin and the remainder mostly of asteroidal origin.

1.42 Density

Estimates of the density of the cometary meteoroids vary from 0.05 to 3.5 g/cm^3, whereas for asteroidal meteoroids the estimates range from 3.5 to 8.0 g/cm^3.

1.43 Velocities

The estimates** range from 11 to 72 km/sec with an average between 28 and 70 km/sec.

1.44 Size

The size of meteoroids varies from about 1 micron to about 500 miles across. A collection of data in graphical form is given in Fig. 70 of the flux (number of particles crossing 1 sq. ft. per day) as a function of meteoroid mass in grams. Data for Fig. 70 were taken from "Meteoroid Protection for Space Radiators," I. J. Loeffler, S. Lieblein and N. Clough, ARS Power Systems Conf., September 1962, Paper No. 2542-63. It has been compiled from various sources and, in addition to those cited above, the following provide much of the original data:

"Rocket, Satellite and Space-Probe Measurements of Interplanetary Dust." IGY Bull. No. 61 (July 1962).

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*See for example:
"Meteor Astronomy," Lovell, A. C. B., Oxford Univ. Press (1954).
"Meteoroid Shielding for Space Vehicles," Rodriguez, D. Aerospace Eng. 19 (12), p. 20 and 53 (December 1960).
"Behavior of Materials in Space Environment," Jaffe, L. D. Rittenhouse, J. B. Tech. Rep. 32-150, Jet Propulsion Lab. (November 1961).
"Space Debris Hazard Evaluation," Davison, E. H. and Winslow, P.C. NASA TN D-1105 (1961).
"Meteoroid Protection for Space Radiators," Loeffler, I. J. Lieblein, S. and Clough, N. NASA Amer. Rocket Soc. Space Power Systems Conf. (Sept. 1962).
**See above references and "On the Velocity Distribution of Meteoroids," Fralko, Y. I. U.S.S.R. Office of Tech. Services, U.S. Dept. of Commerce.

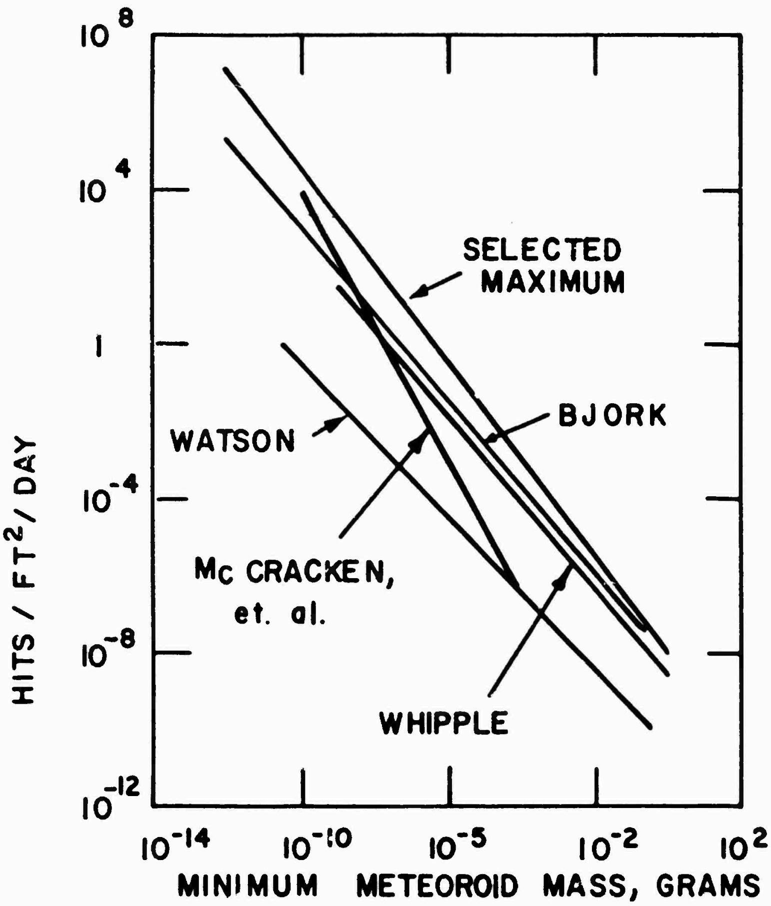

Fig. 70 - Meteoroid mass versus frequency distribution. (Data from Loeffler, Lieblein and Clough, ARS Power Systems Conf., September 1962, Paper No. 2543-62.)

"Particulate Contents of Space." Whipple, F. L. Publ. in "Medical and Biological Aspects of the Energies of Space." Edited by P. A. Campbell, Columbia Univ. Press, New York (1961).

"Relation Between Asteroids, Fireballs and Meteorites." Hawkins, G. S. Astron. J. 64, 450 (1959).

"Density and Mass Distribution of Meteoritic Bodies in the Neighborhood of the Earth's Orbit." Brown, H. J. Geophys. Res. 65, 1679 (1960) and 66, 1316 (1961).

"Direct Measurements of Interplanetary Dust Particles in the Vicinity of the Earth." McCracken, C. W. Alexander W. M. and Dubin, M. Nature 192, 441 (1961).

NOTE: The Venus voyage of Mariner II has provided some additional data on interplanetary flux of micrometeoroids. The Mariner II sent back word of only two impacts during its voyage of particles larger than 1.3×10^{-9} grams, a flux rate of 1/10,000 of that near the earth. (James, J. N. Scientific American, July 1963, pg. 81.)

VII.2 INTER-RELATION BETWEEN COMPONENT DESIGN AND SPACE VEHICLE ENVIRONMENT

2.1 INFLUENCE OF ALTITUDE, AMBIENT GAS DENSITY AND SPACE VEHICLE REQUIREMENTS

Since the liquefier or reliquefier systems will have to operate in high altitude environments in which the ambient gas pressure may be lower than 10^{-9} mm Hg., the systems must be enclosed and high-vacuum sealed to avoid loss of gas.

All components and mounting for all components must be designed to withstand the high-g conditions and vibrations associated with space vehicle launching.

Since very low or zero-g conditions will be obtained in the space vehicles after launch and since they may operate in various attitudes, the systems must be designed to obviate the necessity for any process dependent on the existence of a finite g value. In general this means that single phase refrigerative systems are to be preferred over two-phase systems, since single phase (vapor) systems are unaffected by zero-g environment. Any lubrication process, if used, for refrigeration components must be designed to be without oil, oil-sumps, etc., since this would impose serious design difficulties for zero-g operation. The objective must be to employ oil-free compressors, expanders and other moving mechanisms. This is a problem area, since most large size compressors in use today for liquefaction and reliquefaction use oil lubrication and many employ oil-sumps.

2.2 INFLUENCE OF RADIATIONS AND PLASMAS EXTERNAL TO THE SPACE VEHICLE

Here one is concerned with the possible effects of solar wind, stationary gas, solar flares, and cosmic radiation on the liquefaction or reliquefaction systems. The possible effects on radiators have been studied by Silvern*, who concluded that the erosion of surfaces by these causes is negligible for thicknesses of materials such as is required for minimization of meteoroid damage. Since all pneumatic components of liquefier or reliquefier systems must be of thickness sufficient to obviate meteoroid damage or since they must alternatively be shielded by material of this necessary thickness, it is concluded that erosion effects due to these radiations and/or plasmas is negligible for all pneumatic components. Other components, for example control components (especially electronic and solid-state electronic controls), electric wiring, etc., should be suitably protected with covering against erosion by this source.

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*Silvern, D. H. "Study of Heat Rejection in Space." ABMA Report No. 250-IR-1.

Effects due to these causes other than erosion must also be considered, which fall into the category of "radiation damage." Polymers, organics, plastic materials, glasses, insulations, semiconductors, oils, etc., are more prone to radiation damage than metals, although the full effects of radiation damage at very low temperatures in metals have not yet been fully explored. Where such materials are used, therefore, particularly in any control components employing electronic circuitry, care must be taken in initial design to provide adequate radiation damage shielding, as may be specified in current engineering handbooks on radiation damage.* In view of the fact that such protection may in some instances involve a significant weight increase in the system, it is desirable to avoid altogether, if possible, the use of such materials prone to radiation damage, e.g., controls where possible should be pneumatic.

These considerations lead also to the conclusion that solid-state, semiconductor, refrigeration, even if it appeared feasible under terrestrial environments, would require extended study in radiation fields before it could be considered adequate for space environment.

2.3 NUCLEAR RADIATION

If a nuclear power plant is used in the space vehicle, then nuclear radiation damage to the liquefying or reliquefying systems must be considered. In this case the radiation field will be considerably more intense than that due to the space environment (neglecting passage through Van Allen belts) and the recommendations made in subsection 2.2 must be carefully followed in the design of the system.

2.4 EFFECTS OF METEOROIDS ON COMPONENTS OTHER THAN RADIATORS

The effects of meteoroids on radiators is discussed in some detail in subsection 2.5 below. For this consideration of possible meteoroid damage to the other components of liquefying or reliquefying systems an approximate formulation of the problem is sufficient.

*For example: Nuclear Eng. Handbook, McGraw-Hill (1958)
Radiation Damage in Solids, Billington, D. S. and Crawford, Jr., J. H. Princeton Univ. Press (1961).

Based on data provided by Whipple* and by Bjork**, Rousseau*** gives the following expression for the necessary wall thickness, t, of materials for protection against puncture by meteoroids:

$$t \text{ (inches)} = a\left(\frac{\tau A}{1 - P}\right)^{1/3} \qquad (52)$$

where τ is the mission time in years, A the external exposed surface area and P is the design probability of no critical damage, and where a is dependent on the characteristics of the exposed material, and is equal to 0.001 for steel and 0.0023 for aluminum.

Curves showing t versus exposed area A, taken from Rousseau*** data, are shown in Fig. 71 for steel and aluminum, in which the mission time was assumed to be two years and the design probability of no puncture was assumed equal to 95 per cent. It will be seen from the curves that for steel of wall thickness 0.090", 95 percent protection is afforded for areas of approximately 70 sq. ft. Since in general the components of the liquefying or reliquefying systems, particularly the pneumatic components, must for other reasons have wall thicknesses greater than 0.090" for steel (or the equivalent for other metals), it is concluded that damage to such components (other than the radiator, which is discussed separately below) will be negligible.

2.5 RADIATOR DESIGN

2.51 RADIATOR TEMPERATURE

To assess the optimum temperature for the inlet and outlet fluid to the radiator which serves to dissipate the heat of compression, etc., in space liquefiers and reliquefiers, it is necessary first to estimate the equivalent radiative sink temperature, i.e., the average effective temperature of the space into which the radiator is assumed to be radiating. The equivalent radiative sink temperature, T_r, is the same as the temperature to which the radiator would come in equilibrium if it were not dissipating any of its internal energy.

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*Whipple, F. L. "The Meteorite Risk to Space Vehicles," ARS Journal, October (1957).

**Bjork, R. L. "Effects of Meteoroid Impact on Steel and Aluminum in Space," Rand Corp. Paper No. 1662, December (1958).

***Rousseau, J. Airesearch Mfg. Co., December 1960, ASTIA Document No. 222204.

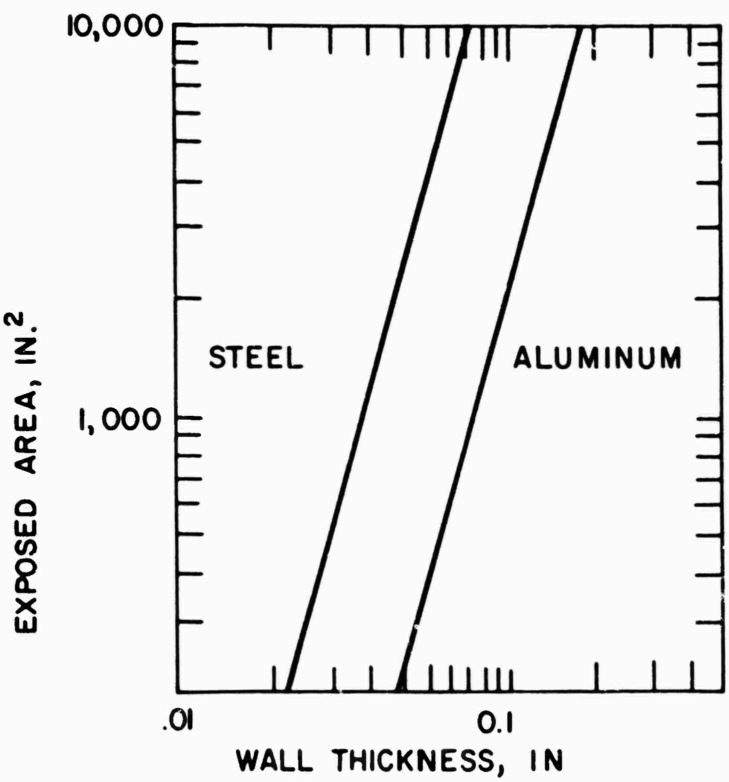

Fig. 71 - Estimated wall thickness for 95% protection against meteoroids versus exposed area for a two year exposure time. (Data from J. Rousseau, Airesearch Mfg. Co., December 1960, ASTIA Document 222204.)

Calculations have been made for T_r by J. Rousseau* who gives for a radiator in the vicinity of the earth:

$$T_r^4 = \frac{\alpha_S C F_S^{\,2}}{\sigma \epsilon \eta_f a F_E \sqrt{1 + \frac{F_S^2}{a^2 F_E^2}}} + \frac{\alpha_S C F_E}{\sigma \epsilon \eta_f \sqrt{1 + \frac{F_S^2}{a^2 F_E^2}}} + \frac{\alpha_T T_E^4 F_E}{\eta_f} \qquad (53)$$

where α_S = radiator coating solar radiation absorptivity.

α_T = radiator coating absorptivity for earth's radiation.

C = solar constant.

F = shape factor (subscript s for sun, E for earth).

a = earth's albedo.

T_E = earth's temperature.

η_f = radiator fin effectiveness.

ϵ = radiator surface emissivity.

σ = Stefen-Boltzman constant

Two radiator configurations were considered as typical examples and the evaluations for T_r as a function of altitude above the earth are given in Fig. 72 (data from J. Rousseau, loc. cit.). For these data Rousseau assumed the following radiator configurations:

1. A flat rectangular plate rejecting heat from one side only, looking down at the earth, with the direct and reflected radiation from the sun at a maximum value.

2. A cylindrical radiator with axis passing through the center of the earth, with direct and reflected radiation from the sun at a maximum value.

In addition, it was assumed that: $\alpha_S = 0.14$; $\alpha_T = 0.9$; $\epsilon = 0.9$; $a = 0.8$; and $\eta_f = 1.0$. The assumptions for α_S and α_T are reasonable figures as may be seen from Table 34, which gives the emissivity (α_T) at 325°K and 675°K and the absorptivity (α_S) for solar radiation for various coating materials. (Data taken from J. Rousseau, loc. cit.)

*Rousseau, J. "Infrared Detector Cooling in Ultrahigh Altitude Vehicles," Airesearch Mfg. Co., December 1960, ASTIA Document No. 222204.

TABLE 34

SOME DATA ON EMISSIVITIES AND ABSORPTIVITIES OF VARIOUS SUBSTANCES

(For source, see text)

Substance	α_T at 325°K	α_s
Al_2O_3	0.98	0.16
CaO	0.96	0.15
ZrO_2	0.95	0.14
MgO	0.97	0.14
$PbCO_3$	0.89	0.12

From Fig. 72 it will be seen that for the plane radiator the equivalent radiative sink temperature, T_r, under conditions of maximum incident radiation varies from about 270°K at an altitude of 200 miles to about 250°K at 100,000 miles, whereas the values of T_r for the cylindrical radiator vary from about 240°K at 200 miles to about 185°K at 100,000 miles.

It is considered of value to extend these estimates of T_r to conditions in space far removed from planets and at distances from the sun between the orbits of Mars and Venus. The computation is as follows.

Radiative energy received an area δA normal to the incident radiation from the sun's surface of radius R at a distance d is given by (for $d \gg R$):

$$\sigma \epsilon_{sun} T_{sun}^4 \delta A \frac{R^2}{d^2 + R^2} \tag{54}$$

where ϵ_{sun} and T_{sun} are the emissivity and surface temperature of the sun and σ is the Stephan-Boltzmann constant.

Let α_s and ϵ be the absorption coefficient and emissivity of the receiving surface; then the equilibrium temperature, T_r, of the receiving surface (assuming it can radiate from both sides) is:

$$T_r = T_{sun} \left(\frac{\epsilon_{sun} \alpha_s}{2\epsilon}\right)^{1/4} \left(\frac{R^2}{d^2 + R^2}\right)^{1/4} \tag{55}$$

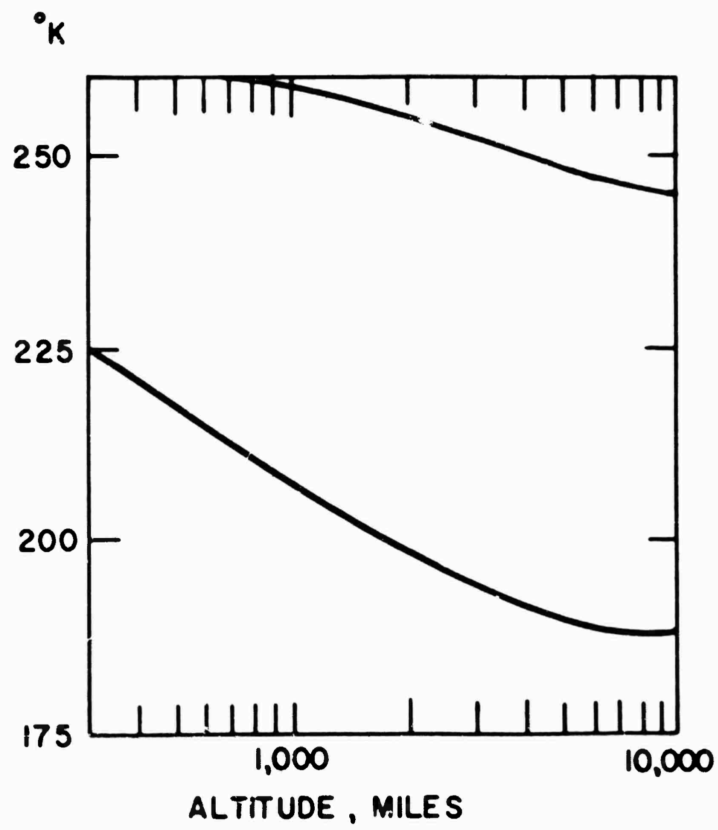

Fig. 72 - Radiative equivalent sink temperature, T_r, versus altitude above earth. (Data from J. Rousseau, Airesearch Mfg. Co., December 1960, ASTIA Document No. 222204.)

T_r then is the equivalent radiative sink temperature for a flat radiator with plane normal to incident radiation. Putting in the values: $\epsilon_{sun} = 1$; $a_s = 0.14$; $\epsilon = 0.9$; R (sun) = 6.96×10^{10} cm. = 4.66×10^{-3} A.U.; $T_{sun} = 6300°K$, we get:

$$T_s \approx 225/\sqrt{d} \ °K \tag{56}$$

where d is in A.U.

Values of T_r for various values of d under the above conditions of maximum incident solar radiation and negligible planetary reflected radiation are given in Table 35.

TABLE 35

EQUIVALENT RADIATIVE SINK TEMPERATURE FOR PLANE RADIATOR (MAXIMUM) IN SOLAR RADIATION FIELD ONLY, ASSUMING $\alpha_s = 0.14$ and $\epsilon = 0.9$

d(A.U.)	Equivalent Planetary Orbit	T_r(°K)
0.723	Venus	264
1.0	Earth	225
1.52	Mars	183

For a cylindrical radiator the corresponding values of T_r are approximately 60% less than the above figures. The exact value of T_r will be a complex function of the radiator geometry and altitude and reference is made to previous work in this field.* For the purposes of this study the significance of the above data and calculations is the provision of working limits for the value of the equivalent radiative sink temperature for the space vehicle liquefaction program covered in this report. From Fig. 72 and the calculations above, T_r can range from about 120°K for a cylindrical radiator distant from the sun at approximately 1.5 A.U. (Mars orbit) to over 270°K for a plane radiator in the vicinity of Venus. The lower limit here is for the case where no attempt is made to minimize the incidence of solar radiation by attitude control; indeed the reverse situation is assumed, namely maximum solar incident radiation. By

*"Infrared Detector Cooling Ultrahigh Altitude Vehicles," Rousseau, J. Airesearch Mfg. Co., December 1960, ASTIA Document No. 222204

suitable geometrical design of the radiator and/or attitude control, values of T_r significantly lower than the above quoted could, if desirable, be obtained.

The surface areas required in the radiator for a given power radiated, P, are a function of the equivalent sink temperature, T_r, and of the inlet temperature, T_1, outlet temperature, T_2, of the fluid passing through the radiator. The expression* for the radiator area is:

$$\frac{A}{P} = \frac{1}{2\sigma\eta_f\epsilon T_r^3(T_1 - T_2)}\left[\tan^{-1}\frac{T_2}{T_s} - \tan^{-1}\frac{T_1}{T_r} + \frac{1}{2}\ln\frac{(T_1 - T_r)}{(T_2 - T_r)}\frac{(T_2 + T_r)}{(T_1 + T_r)}\right] \quad (57)$$

where σ is the Stephan-Boltzmann constant and the other symbols are as already defined.

Fig. 73 shows an evaluation of $\eta_f A/P$ in square feet per kilowatt as a function of T_r for various values of T_1 and T_2. In these evaluations ϵ was taken equal to 0.9, a figure compatible with data of Table 34.

From the curves of Fig. 73 it will be seen that for high inlet temperatures T_1 of about 500°K, the required surface area per kilowatt dissipated is almost independent of T_r in the range T_r = 0°K to 250°K and moreover is only slightly dependent on the temperature difference $(T_1 - T_2)$ between inlet and outlet streams. From a point of view of radiator design, therefore, it is desirable to have T_1 as high as possible. This, however, is extremely undesirable from a point of view of liquefier or reliquefier efficiency, where it is advantageous to reduce the temperature of the liquefier heat sink to as low a value as possible. A compromise, therefore, is necessary, as is discussed in Section VIII.

It will be further noted from Fig. 73 that as the inlet temperature, T_1, is reduced, the required surface area per kilowatt dissipated becomes more strongly dependent on the temperature difference $T_1 - T_2$, larger values of $T_1 - T_2$ requiring much larger values of $\eta_f A/P$.

*"Infrared Detector Cooling in Ultrahigh Altitude Vehicles," Rousseau, J. Airesearch Mfg. Co., December 1960, ASTIA Document No. 222204.

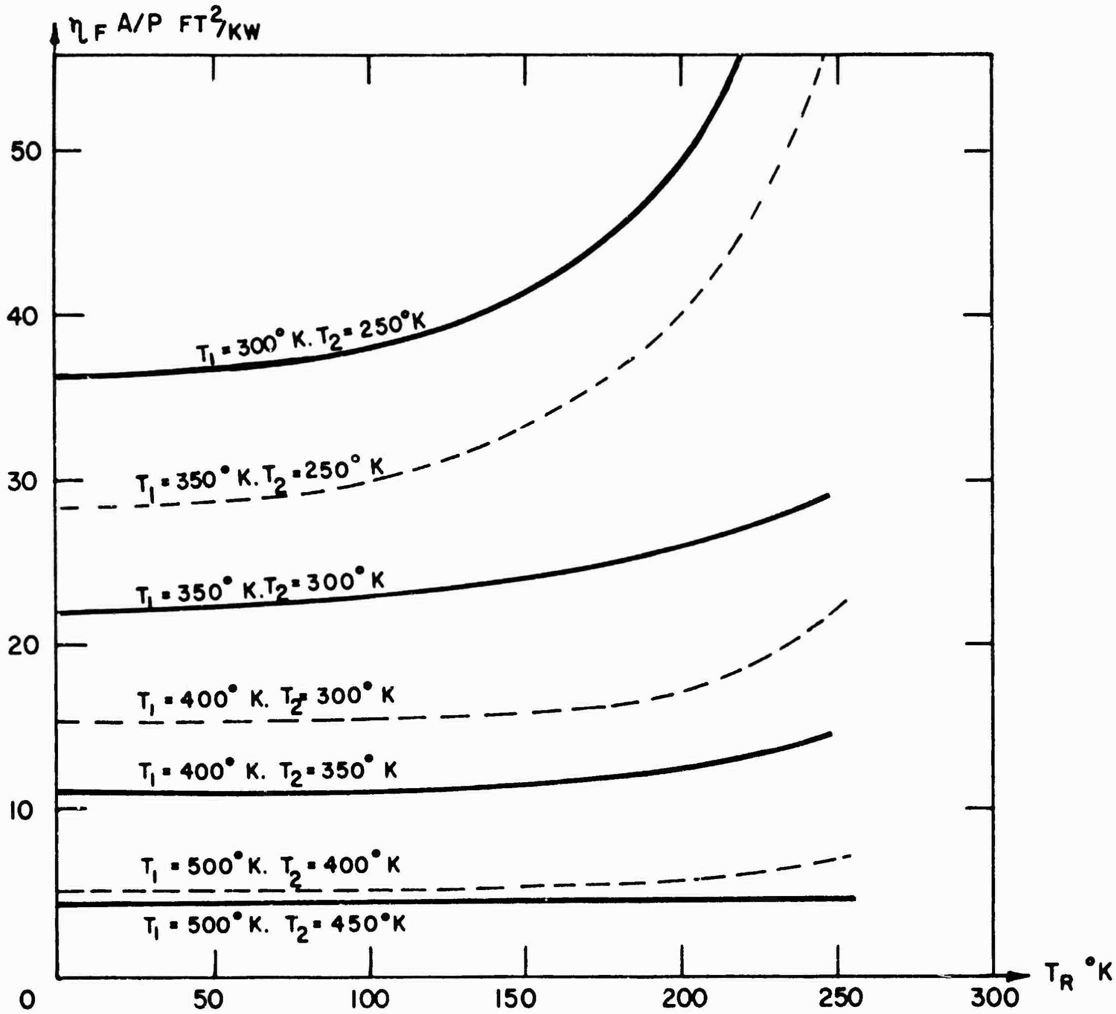

Fig. 73 - *Effective radiator area per kilowatt dissipated ($\eta_f A/P$) in sq. ft., versus equivalent radiative sink temperature, T_r, for various fluid inlet (T_1) and outlet (T_2) temperatures to the radiator. (For sources of data, see text.)*

2.52 RADIATOR PROTECTION FROM METEOROID DAMAGE

As stated in a recent report by Loeffler, Lieblein and Clough*, there are still considerable uncertainty in the design requirements for protection of radiators against impact damage. These authors suggest that the major approaches to be followed are 1) to obtain a better definition of the hazards and the damage mechanisms, 2) to determine ways to reduce the vulnerability of radiators which will result in minimum weight and maximum reliability. These problems have been and continue to be the subject of extensive theoretical and experimental study. References to some of this work have been given above and other pertinent work is enumerated below.** In Section VII.1 above an outline of current information on the meteoroid hazard has been presented. Clearly much remains to be done in this area.

In considering the question of minimization of the vulnerability of space radiators the following approaches are to be considered, as has been discussed, for example, by Loeffler, Lieblein and Clough.

2.521 REDUCTION OF VULNERABLE AREA. By suitable choice of geometry of tube and fin areas in the radiator a minimization of the vulnerable tube area can be achieved.*** Moreover, as is evident from the considerations put forward in Section VII.2.51 above, by increase of the inlet temperature of the fluid to the radiator a reduction of the required area can be obtained. However, this is undesirable from the point of view of liquefier or reliquefier efficiency and a compromise must be reached (see Section VIII).

2.522 DEVELOPMENT OF NON-FLUID RADIATORS. Non-fluid radiators using continuously moving belt or rotating disk surfaces have been proposed and promise significant weight savings. At the present time, however, it is considered premature to base design studies for liquefier or reliquefier systems for the period 1965-70 on these new concepts. In any subsequent updating

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*"Meteoroid Protection for Space Radiators," ARS Power Systems Conf., September 1962, Paper 2542-63.

**See, for example, references above and: Whipple, F. L, "The Meteorite Risk to Space Vehicles," ART Journal, October 1957.

Bjork, R. L. "Effects of Meteoroid Impact on Steel and Aluminum in Space," Rand Corp., Paper No. 1662, December 1958.

"Space Radiator Study," Thompson-Ramo-Wooldridge, ASK Tech. Rep. 61-697, April 1962.

Krebs, R. P. Winch, J. M. and Lieblein, S. "Analysis of a Megwatt Level Direct Condenser-Radiator," ARS Space Power Systems Conf., September 1962, Paper No. 2545-62.

***See, for example, Loeffler, Lieblein and Clough, loc. cit., and Mackay, D. B. "Space Radiator Analysis and Design," July 1962.

of this report, however, notice should be taken of any practical breakthrough in this area.

2.523 SELF-SEALING OR SELF-REPAIRING RADIATORS. A satisfactory practical method of self-sealing or self-repairing has not yet been evolved. Here, therefore, the recommendations made for item 2 above also apply.

2.524 ORIENTATION. Because of the known directional properties of meteoroid movement, choice of favorable orientation of radiators may lead to significant reduction of damage probability or allow weight savings to be effected. Since, however, to date no detailed design information is available, this approach must be the subject of further study.

2.525 ISOLATABLE SEGMENTATION OF RADIATORS. This system envisages the isolation of damaged segments of the radiator and the consequent operation of the system under correspondingly reduced power. It appears that this approach looks promising for large radiators designed for high power (megawatt) levels.

2.526 THICK RADIATOR FLUID PASSAGES WITH OR WITHOUT SHIELDS AND BUMPERS. This is the current approach to the meteoroid damage problem and is the one on which design considerations in this report are based.

According to Loeffler, Lieblein and Clough the desired "armor" thickness, t, for radiator tubing is given by:

$$t \text{ (inches)} \approx \frac{2a}{2.54} \left(\frac{6}{\ }\right)^{1/3} \rho_P^{-1/3} \left(\frac{62.4\rho_P}{\rho_t}\right)^{1/2} \left(\frac{\overline{V}}{c}\right)^{2/3} \left(\frac{\alpha A \tau}{-\ln P}\right)^{1/10} \tag{58}$$

where a = thin plate and space (1.75).

ρ_P = meteoroid particle density (2.7 g/cm^3).

ρ_t = radiator material density (lbs/ft^3).

$\overline{V}$ = average meteoroid velocity (98,400 ft/sec).

$c = 12\sqrt{\frac{E_t g}{\rho_t}}$ where E_t is Young's modulus in lbs/in^2.

$\alpha = 2.54 \times 10^{-9}$.

A = external exposed surface area of tubes (ft^2).

τ = mission time in days.

P = design probability of no critical damage.

2.53 RADIATOR SIZE AND WEIGHT

For a radiator design using tubes, fins and armor as proposed by Loeffler, Lieblein and Clough, these authors present a weight versus surface area similar to that shown in Fig. 74. It will be seen that the weight per square foot of the armored radiator increases with increasing total area. For the radiator size as a function of power dissipated, equivalent radiative temperature, etc., see curves presented in Fig. 73.

VIII. COMPARISON OF CYCLES

VIII.1 GENERAL CYCLE COMPARISON

Before going into a quantitative comparison of cycles for liquefaction and refrigeration, it might be well to discuss qualitatively the bases to be used for comparison. Basically, for our considerations, there are four important categories of comparison to make between different cycles. They are (1) power consumption, (2) weight and volume, (3) reliability, and (4) adaptibility to space environment..

1.1 POWER CONSUMPTION

Thermodynamically, the power consumption for any cycle of refrigeration or liquefaction may be broken into two parts. The first part is the minimum reversible work required to do the job, either liquefaction or refrigeration. (See Section IV.) This minimum reversible work is the same for any two cycles designed to do the same job; for example, two cycles each designed to take 10 lbs/hr of oxygen at 70°F and 1 atm. and convert it to saturated liquid at 1 atm. Assuming the input and output pressures are the same for any two cycles doing the same job, their power consumptions will be different because one cycle contains more irreversibilities than the other. The thermodynamic quantity which is a measure of irreversibility is the entropy, usually designated by the letter S. For any cycle, the total work equals the reversible work, plus the irreversibility in the compressor, plus $T_O \Sigma\Delta S$ for the rest of the system, where

T_O = ambient temperature or temperature of heat rejection,

Δ_S = entropy change of working fluid in any section of the cycle outside the compressor.

It is roughly correct to assume the compressor irreversibility to be directly proportional to total power. Making that assumption, we then see that for any two cycles doing the same job, working at the same pressures and temperatures, the one with the greater $\Sigma\Delta S$ will consume more power. For any cycle, no matter how complicated, the irreversibilities are due to five fundamental processes: (1) Heat transfer processes, (2) Pressure drops through piping and heat exchangers, (3) Non-isentropic expansion, (4) Heat leak from the outside universe, and (5) Heat conduction from the warmer to the colder parts of the system.

It is valuable, moreover, to differentiate between irreversibilities inherent in the cycle and irreversibilities caused by components which are less than 100 per cent efficient. The magnitude of the pressure drops is largely a matter of equipment

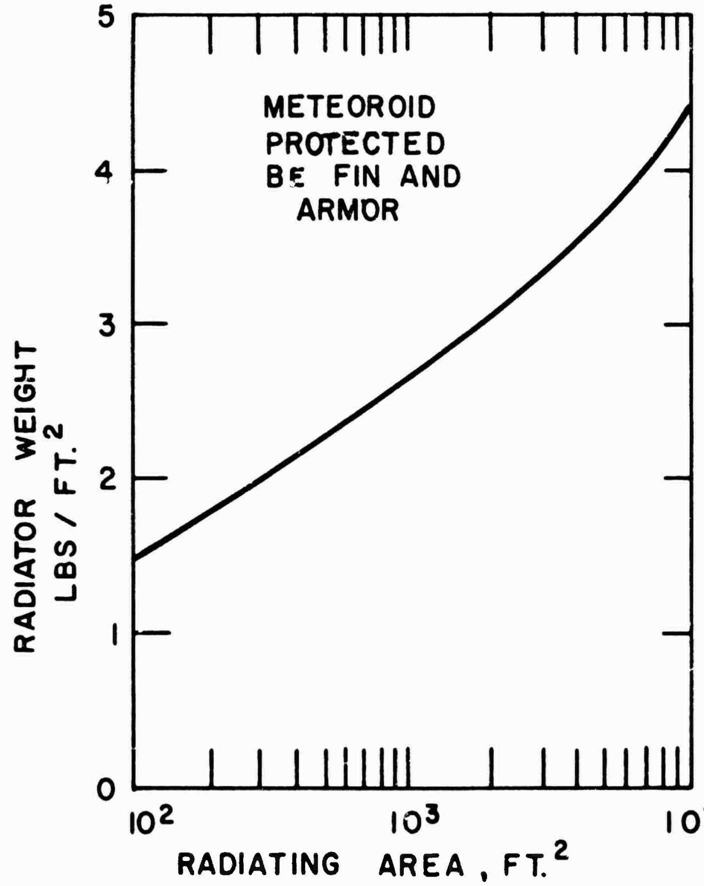

Fig. 74 - Radiator weight as a function of area. (Data from Loeffler, Lieblein and Clough, ARS Space Power Systems Conf., September 1962, Paper No. 2543-62.)

design and not fundamentally inherent in the cycle. Temperature differences in heat exchangers are, in general, partly due to heat exchanger inefficiency and partly due to the cycle itself. Any cycle in which heat is transferred between fluid streams of unequal heat capacity has inherent in it a thermo-dynamic irreversibility. This is an important point, because it tells us, for example, that there is a significant irreversibility in the main heat exchanger of a Linde-type nitrogen liquefier no matter how efficient the exchanger may be. The realization of this fact leads directly to the precooled Linde cycle or to the Claude cycle, in both of which extra refrigeration is applied to the high-pressure stream in the main exchanger. The most extremely undesirable heat transfer, from a power standpoint, is to exchange latent heat with sensible heat. It is this which accounts for the relative power disadvantage, for reliquefaction purposes, of the dense-gas cycle as compared to the Stirling cycle or the Taconis cycle.

A basic irreversibility is inherent in any cycle which employs Joule-Thomson expansion. Expansion through an engine or turbine, on the other hand, approaches a constant-entropy operation, falling short only by virtue of inefficiencies in the expander. However, as may be seen on a T-S diagram, isenthalpic expansion near the critical conditions is quite efficient; in fact it has in this region a practical efficiency approximately as good as an expansion engine. Therefore, a valve would be chosen over an engine for expansion near critical conditions. A valve is also usually chosen for expansion into the liquid region, although it is possible to expand into the liquid region with a turbo-expander. For most cases of gas expansion, an expansion engine has the advantage of decreasing the power consumption of the system, while the valve has the advantages of reliability and low weight. In choosing a means of expansion a compromise must in general be made between the reliability and low weight of the valve and the power savings obtained by use of an engine or turbine.

It should be pointed out that inefficiency in the components does not necessarily imply faulty design. A perfect heat exchanger would combine an infinite heat transfer surface with infinitesimal pressure drop. A perfect expander would operate without friction. These ideals are unattainable, even with the best possible engineering designs.

1.2 WEIGHT AND VOLUME

For space-borne application, the relative weights and volumes of cycles under consideration will certainly be a factor in choosing an optimum one. For the purposes of this study, the optimum cycle in terms of weight, volume and power consumption would be that which has the minimum total weight and volume

of system and power supply. However, in fact, these parameters are not in themselves sufficient to determine the optimum cycle. Consideration must also be taken of the weight and area of the space radiator used as a heat sink for the refrigeration or liquefaction cycle. There is no theoretical approach to determining system weights or volumes. For any cycle these are empirical functions of such parameters as capacity and temperature level; however, as will be seen below, cycles with larger power requirements also have larger weights and volumes.

1.3 RELIABILITY (EXCLUDING SPACE RADIATORS)

The major causes of failure in cryogenic systems are failure of moving parts and the effects of contamination in the working fluid. In terrestrial liquefaction processes, the gas to be liquefied may contain contaminants which can freeze in various parts of the cycle. In a space vehicle, great care must be taken to ensure highest purity of the fluids with which a refrigerator, reliquefier, or liquefier is initially charged and there should be no source of contamination due to compressor lubricants. It is desirable, therefore, to avoid any compressor contaminant. The major moving parts which would be reliability factors are compressors, expanders, and switching valves. In the case of compressors, reliability generally decreases with increased discharge pressure. This is because of the higher stresses on equipment from high pressures and also because reciprocating compressors are often used for high compression ratios. Reciprocating compressors are less reliable than turbines, largely because of the valves which must switch open and shut on the intake and exhaust portions of each compression stroke. As for expansion, an engine is clearly less reliable than a valve. It is generally considered that turbines for compression and expansion are more reliable than reciprocating machinery. However, they are less capable of being miniaturized than reciprocating machines. Cycles, such as the Stirling cycle, which employ no valves can be made highly reliable even though they employ reciprocating machinery. The third mentioned reliability factor is switching valves. It would seem desirable to include manual by-pass valves in parallel with all switch valves, so that in the event of failure of the latter, the cycle can continue to be operable. The redundancy would add little weight to the total system and would insure continued operation. However, where possible, cycles should be chosen which do not require such valves.

1.4 ADAPTABILITY TO SPACE ENVIRONMENT

The space environment will differ from the terrestrial in three main ways: (1) There will probably be no gravity; (2) There will be large acceleration and vibration forces, particularly at take-off and landing; and (3) Effective ambient temperatures may range from about 350°K or warmer down to cryogenic temperatures (See Section VII).

The lack of gravity will have a large effect on those cycles which depend on heat transfer from a boiling liquid to provide refrigeration. Efficiency of boiling heat transfer is greatly increased by convection currents in the boiling liquid. Since these will be absent in the absence of gravity, it will probably be necessary to provide mechanical agitation or circulation for a boiling liquid in space.

Susceptibility to acceleration and vibration forces is hard to predict with any accuracy. This is generally a subject for experimental investigation. In general, however, one might expect that cycles contained in one compact unit, as for example the Stirling cycle, would be preferable to those which consist of several components piped together.

The possibility of ambient temperatures ranging from terrestrial ambient down to very low temperatures imposes problems in cycle design and in cycle choice in addition to any which are involved in space radiator engineering. Matters concerning space-radiators have been discussed in Section VII.

Although the general space vehicle design will largely determine and possibly restrict the range of ambient temperature of the components and systems mounted therein, consideration must be given to the question of component operation over a wide range of ambient temperature. These considerations must include mechanical as well as thermodynamic design problems. The major mechanical problem that would be encountered would be in the design of compressors. Clearly the development of such compressors in the future is a major requirement.

If one assumes that compressors or combined compressors and expanders, as in the Stirling cycle, can be developed which operate satisfactorily in space vehicles, the major mechanical problems involved therein would involve the development of suitable bearings, gears, piston rings, etc., and their lubrication, where necessary, and the possible binding or excessively loose fits of moving parts occurring when changes of the ambient temperature occurs. It is satisfactory to note that in a few instances, as for example miniature Stirling cycles,* these problems have already been solved.

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*For example the Malaker Laboratories Inc. "Cryomite" refrigerators.

1.5 TABULAR SUMMARY OF GENERAL CYCLE COMPARISON

To conclude this general discussion of the basic principles of cycle comparison, the following tables (Tables 36 and 37) present significant items for consideration other than efficiency, weight, and volume. These tables present a compilation of (1) the main causes of inherent irreversibility, (2) a listing of the heavy components (excluding the main compressor, the space radiator and the load cooler), (3) the location of the moving parts (excluding the main compressor) and (4) the major disadvantages in space environment for the nine major significant cycles which have been considered in Section V. Consideration of the relative efficiencies, weights, and volumes of the various cycles is made below in subsection VIII.2.

TABLE 36

QUALITATIVE CYCLE COMPARISON (1)

	Cascade Compressed Vapor	Single Stage J-T System without Precooling	Single Stage J-T System without Precooling	Two-Stage J-T Process
Main Causes of Inherent Irreversibility	J-T expansion. Heat transfer at unequal heat capacity	J-T expansion. Heat transfer at unequal heat capacity	J-T expansion	J-T expansion
Listing of Heavy Components (Excluding Comp., Space Radiator, Load cooler)	Extra Compressors. Multiplicity of Heat Exchangers	Main Heat Exchanger	Precooler. Main Heat Exchanger	Main Heat Exchanger
Location of Moving Parts (Excl. Comp.)	Extra Compressors in Addition to Main Compressor	None	None	None
Major Disadvantages in Space Environment.	Two phases in working fluid. Large number of components	Two phases in working fluid	Two phases in working fluid	Two phases in working fluid

TABLE 37

QUALITATIVE CYCLE COMPARISON (2)

	Systems Employing One Expansion Engine Only	Systems Employing Multi-Stage Expansion Engines Only	Systems Employing Expansion Engines Plus J-T Expansion	Stirling Cycle	Taconis Cycle
Main Causes of Inherent Irreversibility	Heat transfer at unequal heat capacity. Transfer of latent for sensible heat (reliquefaction).	Transfer of latent for sensible heat (reliquefaction).	J-T Expansion.	None	None
Listing of Heavy Components (Excl. Compressor, Space Radiator, Load Cooler)	Expander, Heat Exchangers.	Expanders, Heat Exchangers.	Expander(s), Heat Exchangers.	Regenerator. Expansion Piston.	Regenerator. Expansion Piston.
Location of Moving Parts (Excl. Compressor)	Expander.	Expanders.	Expander(s).	Expansion Piston.	Expansion Piston. Switch Valves.
Major Disadvantages in Space Environment	None	Large number of components.	Two phases in working fluid. Large number of components.	None	None

VIII.2 QUANTITATIVE COMPARISON OF CYCLES

Tables 38 through 51 present estimates based on the considerations put forward in this report of many significant quantitative factors which must be considered, in addition to those discussed in subsection VIII.1 above (and tabulated in Tables 36 and 37), when making a comparison of cycles. The quantitative factors presented in the ensuing tables are: (1) coefficient of performance (for refrigerators or reliquefiers) or kW-hr/lb required for liquefaction, (2) total required input power for reliquefaction or liquefaction at either 1000 lbs/day or 100 lbs/day, (3) compressor and motor weight (lbs), (4) compressor, motor and liquefier weight (lbs), (5) radiator weight (lbs), (6) total weight of compressor, motor, liquefier and radiator (lbs), (7) power generation subsystem weight (lbs), (8) Radiator area (sq. ft.). (9) Compressor volume. (cu. ft.) and (9) Compressor and liquefier volume. (cu. ft.)

These factors are tabulated for various cycles for operation down to 77°K, including· (1) Single stage J-T, (2) Single stage J-T with precooling, (3) two stage J-T without precooling, (4) Two stage J-T with precooling, (5) Systems with one expander (6) Stirling cycle (single stage), (7) Gifford-McMahon engine, (8) Claude cycle (single expander). For operation to lower temperatures, for hydrogen and helium liquefaction and reliquefaction, the following cycles have been quantitatively considered in these tables: (9) Single stage J-T with precooling (10) systems with one expander, (11) compound systems. (12) Stirling cycles in conjunction with J-T circuits.

The tables cover the following liquefaction and/or reliquefaction systems:

Table	Title
38	Reliquefaction of N_2 at 1000 lbs/day
39	Reliquefaction of F_2 at 1000 lbs/day
40	Reliquefaction of O_2 at 1000 lbs/day
41	Reliquefaction of N_2 at 100 lbs/day
42	Reliquefaction of F_2 at 100 lbs/day
43	Reliquefaction of O_2 at 100 lbs/day
44	Liquefaction of N_2 at 1000 lbs/day
45	Liquefaction of F_2 at 1000 lbs/day
46	Liquefaction of O_2 at 1000 lbs/day
47	Liquefaction of N_2 at 100 lbs/day
48	Liquefaction of F_2 at 100 lbs/day
49	Liquefaction of O_2 at 100 lbs/day
50	Reliquefaction of H_2 at 1000(100) lbs/day
51	Reliquefaction of He at 1000(100) lbs/day

In addition Table 52 presents estimates on the kW-hr/lb for liquefaction and the total input power requirements for liquefaction of H_2 and He at 1000 lbs/day for various cycles.

IN USING THESE TABLES, IT IS NECESSARY TO NOTE THE FOLLOWING EXPLICIT COMMENTS WHICH DETAIL THE METHODS BY WHICH THE TABLES HAVE BEEN DRAWN UP AND THE ESTIMATED ACCURACY, RELIABILITY AND/OR APPLICABILITY OF THE DATA.

2.1 COEFFICIENT OF PERFORMANCE AND/OR kW-HR/LB

The detailed quantitative methods whereby the coefficient of performance and/or the kW-hr/lb for liquefaction for each cycle have been estimated are given in Section V. It is to be noted that the evaluations are for the total overall practical performance for which allowance has been made not only for cycle inefficiencies but also for motor and motor-drive inefficiencies and for non-isothermal conditions in compression. For the detailed evaluations of all these items, see Section V. It is also important to note that all the data are obtained assuming that the exit temperature of the compressor aftercooler (radiator) is 300°K.

Two evaluations are given for C.P. or kW-hr/lb, namely: conservative (cons), which is based on current practice and estimated optimistic value (opt. est.) which is based on a reasonable prognostication of future capabilities in the period 1965-1970. For estimates of practical performances of some miniature systems, see subsection VIII.3.

2.2 COMPRESSOR AND MOTOR WEIGHT

The data given in the tables for combined compressor and motor weights are taken from the curves presented in Section VI for reciprocating compressors (Fig. 60). These curves constructed from manufacturers' data on currently available compressors and, as is evident from Fig. 60, there is a considerable range of weights for any given input power. This is understandable since no detailed account has been taken of compressor type, as for example output pressure delivered, cooling medium, etc. Moreover it must be remembered that the compressor data used was for compressors designed with no necessary consideration being made with regard to minimization of weight. Moreover centrifugal compressors have not been taken explicitly in account for lack of data on relatively small scale units. It is evident therefore that there is a considerable area of ignorance here and this has been allowed for in the tables by specifying three different weights, as follows: (a) "conservative upper" (cons. up) which is taken as the highest weight given in Fig. 60

for the "conservative" power requirement. (b) "conservative lower" (cons. low) which is taken as the lowest weight given by Fig. 60 for the "estimated optimistic" power requirement, and (c) an optimistic estimate of weight to be expected for compressors now or in the near future which has been computed at 67% of the "conservative lower" value. It should be noted that no compressor weight evaluations have been given in Tables 38 though 49 for single stage Stirling cycle systems, since in these cycles the compressor, expander and refrigeration system are an integral unit.

The difficulties mentioned above concerning the problem of specification of combined compressor and motor weights are further enhanced in considering H_2 and He reliquefiers. (Tables 50 and 51). With the exception of the cycle "Systems with one expander" for H_2 reliquefaction and possibly the compound systems, the cycles considered have two or more compressors (or Stirling engines). The assessment of the weight here has been arbitrarily made, using Fig. 60, assuming the weight of the compressors and motors to be equivalent to that of one compressor and motor with the same total input power as the sum of the multiple compressor input powers.

As is evident from the above comments and from the tables, there is a wide range for compressor weights and no great claim is made for accuracy in this respect. Since, as is pointed out elsewhere, for space operation special oil-free compressors must be employed or developed, it is likely that the evaluations given here will have to be substantially revised for weight subsequent to this compressor development. The evaluations given here therefore must be taken as a rough guide only.

2.3 COMPRESSOR, MOTOR AND LIQUEFIER WEIGHTS

The estimates given in the tables for the combined compressor and motor weights are discussed immediately above. For all cycles, except the Stirling cycle which does not employ a separate compressor, the complete refrigerator or liquefier system weight is largely determined by the weight of the compressor (with its coolers, filters, etc.) Adequate data on the weights (or volumes) of practical reliquefiers and liquefiers in the ranges considered in this report are not available. In consequence the following approximate assessments were made and have been used in compilation of the tables: (a) for single-stage J-T systems and two-stage J-T systems without precooling the reliquefier or liquefier weight was assessed at one third (1/3) of the "conservative lower" weight of the compressor and motor required for the single-stage J-T system; (b) for single-stage precooled J-T systems and for

two-stage precooled J-T systems for the temperature range down to 77°K, the reliquefier or liquefier weight was assessed at one third (1/3) of the "conservative upper" weight of the compressor and motor required for the single-stage precooled J-T system; (c) for all other systems, except the Stirling cycle, the reliquefier or liquefier weight was assessed at one third (1/3) of the "conservative upper" weight of the compressor(s) and its motor(s). Although this assessment may appear somewhat arbitrary, it is intended to give a rough guide only, since, as noted above, there is considerable possible latitude also in the compressor weight assessment. Three different possibilities are tabulated to cover the range of the possible weights of the compressor, motor and reliquefier (liquefier), namely: (a) "conservative upper" (cons. up) being given by adding the assessed reliquefier (liquefier) weight to the conservative upper evaluation of the compressor-motor weight, (b) "conservative lower" (cons. low), being given by adding the assessed reliquefier (liquefier) weight to the conservative lower compressor-motor weight and (c) an estimated optimistic weight which could be attained by use of improved design and weight-saving materials.

It is not possible to give an accurate assessment from theoretical considerations of the weights and volumes of Stirling cycle liquefiers and reliquefiers for operation down to 77°K at this time. The approach taken has been to assess the weights and volumes of existing machines with their motors which operate in the range of interest to this report. Current estimates of these data made from design data of Malaker laboratories are presented in Figs. 75 and 76 which plot conservative and optimistic evaluations of the volume and weight as a function of overall input power of Stirling cycle liquefiers and reliquefiers (radiators not included) suitable for liquefaction of nitrogen, fluorine and oxygen.

The conservative evaluations are based essentially on estimated current practice and the optimistic evaluations are based on considerations of improved design and of the use of lightweight and advanced metals in their construction. Again it should be emphasized that these evaluations are intended to serve as a rough guide only. The data given in the tables for the Stirling cycles are based on the curves of Figs. 75 and 76.

2.4 RADIATOR AREAS AND WEIGHTS

The radiator areas were obtained from Fig. 73 assuming the cooling fluid inlet temperature to the radiator to be 400°K, the cooling fluid outlet temperature from the radiator 300°K, an equivalent radiative sink temperature, T_r, of 250°K, and a radiator fin effectiveness (f) equal to unity. (See section VII for data). The value of T_r chosen corresponds (see Section VII) to

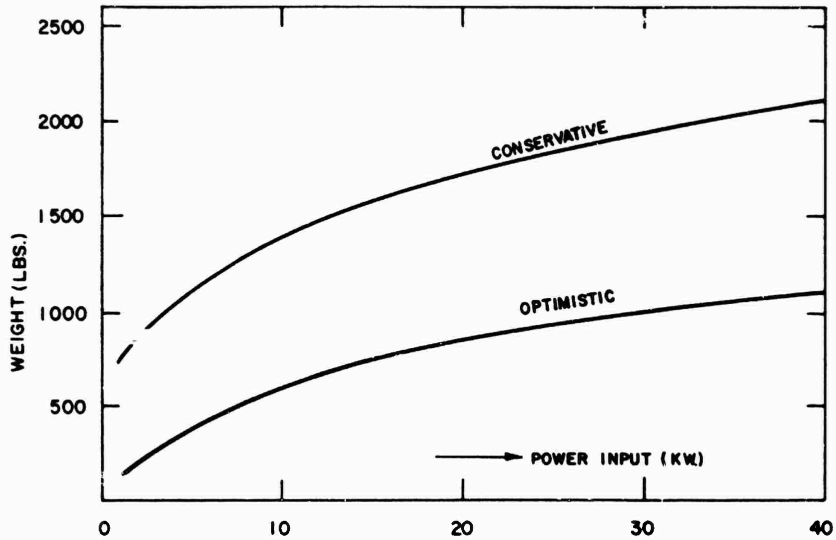

Fig. 75 - Weight of single-stage Stirling cycle system for operation down to 77°K.

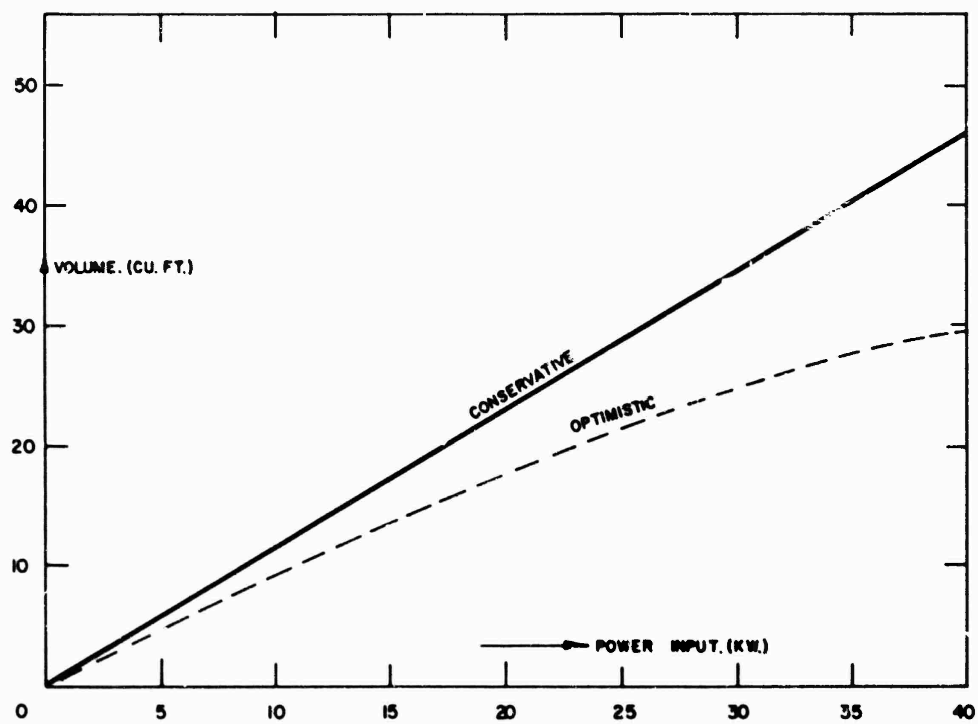

Fig. 76 - Volumes of single-stage Stirling cycle systems for liquefaction or reliquefaction of N_2, F_2 or O_2.

a realizable figure. (Higher values than this would impose greater radiator area requirements, as is shown in Fig. 73). Under these assumptions the area requirement is 22.5 sq. ft. per kilowatt radiated, and this figure has been adopted in the tables. Clearly, under other environmental conditions the radiator area requirements will vary and a compromise between radiator weight and area and reliquefier or liquefier weight must be made. This will be discussed below. In order to simplify our tabulations however, a particular and reasonable value of $\eta A/P$ for the radiator has been assumed; but sufficient data is provided in the report to permit evaluation of radiator parameters under other possible space conditions. Two evaluations for the radiator area are given in the table, both based on $\eta A/P = 22.5$ q. ft./kW, one corresponding to the "conservative" input power requirement of the reliquefier or liquefier, (cons), and one corresponding to the "estimated optimistic" power requirement. (opt).

The radiator weight is taken from Fig. 74 for a Be fin armor protected radiator, which is based on data of Loeffler, Lieblein and Clough (ARS Space Power Systems Conf. Sept. 1962. Paper No. 2543-62). Again two weights are given, based on the "conservative" and "estimated optimistic" radiator areas.

2.5 POWER GENERATION SUBSYSTEM WEIGHT

The data for the power subsystem generation weights are taken from Fig. 77, which was provided by ASD. The authors of this report have made an extrapolation of the original curves to lower power levels, which extrapolation is shown by the broken lines. Fig. 77 shows a range of possible subsystem weights for any given output power. In the tables two evaluations are given for the power generation subsystem weights, a "conservative" value (cons) based on the "conservative" total power requirement and on the highest curve of Fig. 77 and an "optimistic" value (opt.) based on the "estimated optimistic" total power requirement and on the lowest curve of Fig. 77.

2.6 COMPRESSOR AND MOTOR VOLUME

The data given in the tables under compressor volumes are taken from the curves of Fig. 61 (Section VI) for reciprocating compressors. These curves are constructed from manufacturer's data on currently available compressors with their motors and, as is evident from the figure, there is a considerable range of volumes for any given input power. This is understandable since no detailed account has been taken of compressor type, as for example output pressure delivered, cooling medium, etc. Moreover, it must be remembered that the compressor data used was for compressors designed with no necessary restrictions on volume. In addition centrifugal compressor data

are not included. There is here, as in compressor weights, a considerable area of ignorance and this has been allowed for in the tables by specifying three different volumes, as follows: (a) "conservative upper" (con. up) which is taken as the largest volume given in Fig. 61 for the "conservative" power requirement, (b) "conservative lower" (con. low) which is taken as the smallest volume given in Fig. 61 for the "estimated optimistic" power requirement and (c) an optimistic estimate of volume which could be expected for compressors now or in the near future, which has been computed at 67% of the "conservative lower" value.

The difficulties mentioned above concerning the problem of specification of compressor volumes are further enhanced in considering H_2 and He reliquefiers. (Tables 50 and 51). With the exception of the cycle "Systems with one expander" for H_2 reliquefaction and possibly the compound systems, all cycles considered have two or more compressors (or Stirling engines). The assessment of the volume has been arbitrarily made, using Fig. 61, assuming the volume of the compressors and their motors to be equivalent to that of one compressor and motor with the same total input power as the sum of the multiple compressor input powers.

As is evident from the above comments and from the tables, there is a wide range for compressor-motor volumes and no great claim is made for accuracy in this respect. Since, as is pointed out elsewhere, for space operation special oil-free compressors must be employed or developed, it is likely that the evaluations given here will have to be substantially revised subsequent to this compressor development. The evaluations given here, therefore, must be taken as a rough guide only.

2.7 COMPRESSOR, MOTOR AND LIQUEFIER (RELIQUEFIER) VOLUMES

As indicated in the tables, there is little information on this parameter, except, as noted above, for the Stirling cycle. It can be assumed however that for all systems the compressor(s) with motor(s) occupy the major fraction of the total system volume.

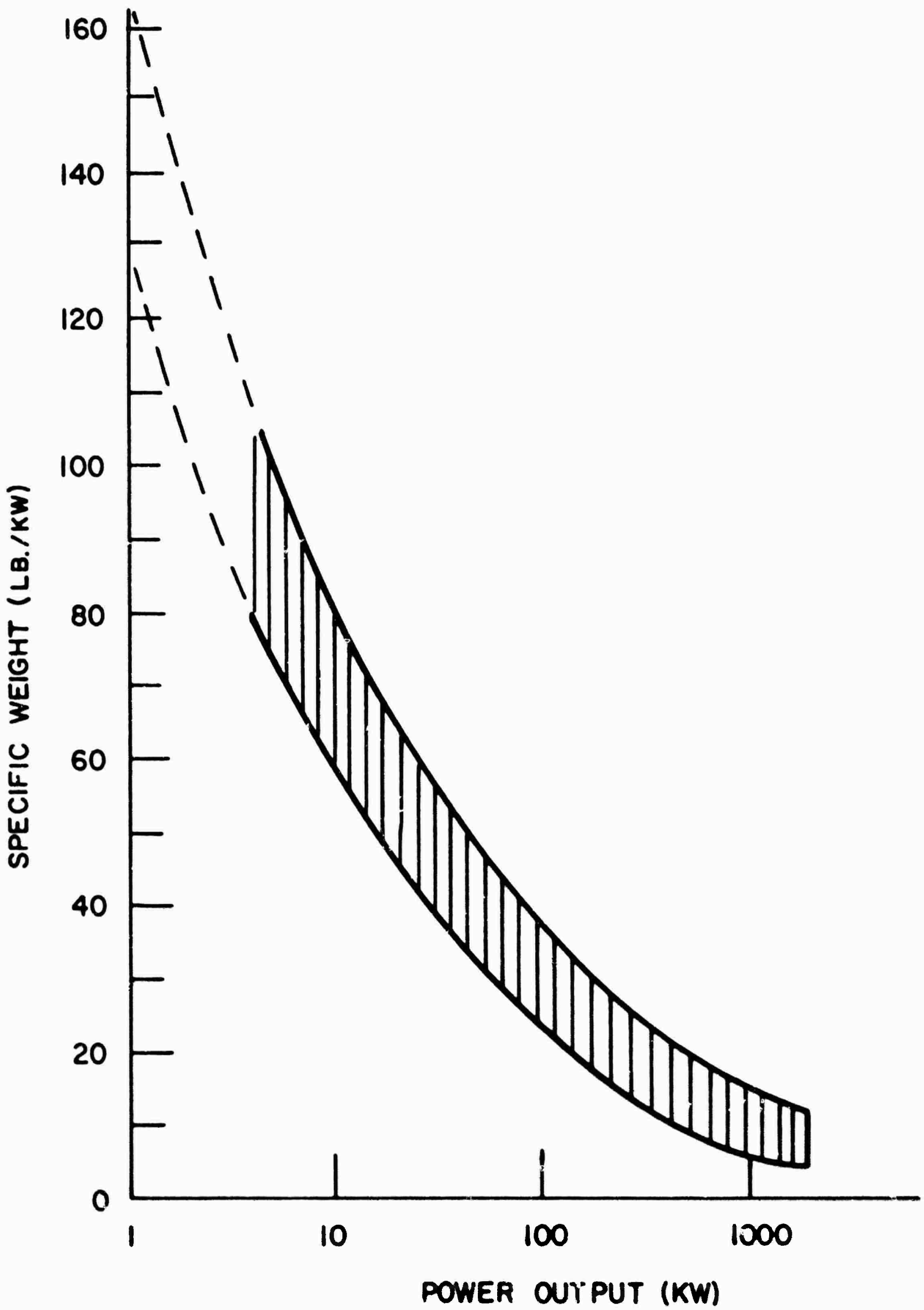

Fig. 77 - Specific weight of anticipated power generation subsystem.

TABLE 38

RELIQUEFACTION OF N_2 AT 1000 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
Coeff. of performance.	Cons. Opt. (Est)	0.0295 0.0325	0.0443 0.0487	0.0562 0.0618	0.0713 0.0784	0.065 0.081	0.084 0.097	0.0675 0.078
Required input power. kW for 1000 lb/day.	Cons. Opt. (Est)	35.6 32.3	23.7 21.6	18.7 17.0	14.7 13.4	16.2 13.0	12.5 10.8	15.6 13.5
Compressor and motor weight. lb.	Cons. (up) Cons. (low) Opt. (Est)	3200 2150 1420	2500 1450 970	2200 1200 800	1850 980 650	1950 950 630	- - -	1900 950 630
Compressor, motor and liquefier weight lbs.	Cons. (Up) Cons. (low) Opt. (Est)	3950 2870 1900	3330 2280 1520	2920 1920 1280	2680 1810 1200	2600 1600 1070	- 1500 620	2530 1580 1050
Radiator weight. lbs.	Cons. Opt.	2000 1700	1200 1050	880 800	660 600	760 590	550 465	730 600
Total weight of compressor, motor liquefier and radiator. lbs.	Cons. (up) Cons. (low) Opt. (Est)	5950 4570 3600	4530 3330 2570	3800 2720 2080	3340 2660 1800	3360 2190 1660	- 2050 1085	3260 2180 1650
Power generation sub-system weight. lbs.	Cons. Opt.	1850 1150	1500 970	1200 780	1000 670	1050 650	870 590	1080 680
Radiator area. sq. ft.	Cons. Opt.	802 728	533 486	421 383	332 302	363 293	282 244	352 304
Compressor Volume cu. ft.	Cons. (up) Cons. (low) Opt. (Est)	135 85 57	85 50 33	65 37 24	50 23 15	55 23 15	- - -	55 23 15
Compressor and liquefier volume. cu. ft.	Cons. (up) Cons. (low) Opt. (Est)	add on approximately 33% to above volumes.					14.5 9.5	

Table 39

RELIQUEFACTION OF F_2 AT 1000 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
Coeff. of performance.	Cons.	0.0346	0.0518	0.0659	0.0835	0.073	0.114	0.0768
	Opt. (Est)	0.0381	0.0570	0.0725	0.0918	0.091	0.131	0.088
Required input power. kW for 1000 lb/day.	Cons.	26.0	17.4	13.7	10.8	12.3	7.9	11.7
	Opt. (Est)	23.4	15.8	12.4	9.8	9.9	6.9	10.2
Compressor and motor weight. lb.	Cons. (up)	2400	1950	1750	1550	1700	-	1650
	Cons. (low)	1550	1050	900	700	740	-	750
	Opt. (Est)	1030	700	600	470	490	-	500
Compressor, motor and liquefier weight lbs.	Cons. (up)	2920	2600	2270	2200	2270	-	2200
	Cons. (low)	2070	1700	1420	1350	1310	1280	1300
	Opt. (Est)	1370	1130	950	900	880	480	870
Radiator weight. lbs.	Cons.	1350	840	620	465	550	310	505
	Opt.	1150	575	560	405	415	250	415
Total weight of compressor, motor liquefier and radiator. lbs.	Cons. (up)	4270	3440	2890	2665	2820	-	2705
	Cons. (low)	3220	2275	1980	1755	1725	1590	1715
	Opt. (Est)	2520	1705	1510	1305	1295	730	1285
Power generation subsystem weight. lbs.	Cons.	1500	1180	970	820	880	720	860
	Opt.	930	760	640	570	580	500	580
Radiator area. sq. ft.	Cons.	585	390	310	245	275	178	265
	Opt.	525	355	280	220	225	155	230
Compressor Volume. cu. ft.	Cons. (up)	95	60	47	36	42	-	40
	Cons. (low)	57	30	23	18	18	-	20
	Opt. (Est)	38	20	15	12	12	-	13
Compressor and liquefier volume. cu. ft.	Cons. (up)	add on approximately 33% to above volumes					-	
	Cons. (low)						9.2	
	Opt. (Est)						6	

TABLE 40

RELIQUEFACTION OF O_2 AT 1000 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
Coeff. of performance.	Cons.	0.0373	0.0559	0.0711	0.0902	0.077	0.133	0.0813
	Opt. (Est)	0.0410	0.0615	0.0782	0.0992	0.096	0.153	0.094
Required input power. kW for 1000 lb/day.	Cons.	30.0	20.1	15.8	12.5	14.6	8.4	13.8
	Opt. (Est)	27.4	18.2	14.3	11.3	11.7	7.4	12.0
Compressor and motor weight. lb.	Cons. (up)	2800	2200	1850	1750	1850	--	1750
	Cons. (low)	1750	1200	1000	900	900	--	950
	Opt. (Est)	1170	800	670	600	600	--	630
Compressor, motor and liquefier weight lbs.	Cons. (up)	3380	2930	2430	2580	2470	--	2330
	Cons. (low)	2330	1930	1580	1630	1520	1300	1530
	Opt. (Est)	1550	1290	1050	1090	1010	500	1020
Radiator weight, lbs.	Cons.	1620	1000	750	545	670	340	670
	Opt.	1400	880	660	470	515	275	520
Total weight of compressor, motor liquefier and radiator. lbs.	Cons. (up)	5000	3930	3180	3125	3140	--	3000
	Cons. (low)	3730	2810	2240	2100	2035	1640	2050
	Opt. (Est)	2950	2170	1710	1560	1525	775	1540
Power generation subsystem weight. lbs.	Cons.	1700	1300	1100	930	1020	760	970
	Opt.	1150	840	720	570	580	520	630
Radiator area. sq. ft.	Cons.	675	455	355	280	330	190	310
	Opt.	615	410	320	255	265	165	270
Compressor Volume cu. ft.	Cons. (up)	110	67	55	48	50	--	47
	Cons. (low)	70	42	25	22	22	--	22
	Opt. (Est)	46	28	16	15	15	--	15
Compressor and liquefier volume. cu. ft.	Cons. (up)	add on approximately 33% to above volumes.						
	Cons. (low)						8.5	
	Opt. (Est)						6.5	

TABLE 41

RELIQUEFACTION OF N_2 AT 100 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
Coeff. of performance.	Cons.	0.0262	0.0393	0.0498	0.0634	0.062	0.075	0.045
	Opt. (Est)	0.0288	0.0432	0.0548	0.0697	0.077	0.086	0.052
Required input power. kW for 1000 lb/day.	Cons.	4.02	2.67	2.11	1.66	1.70	1.40	2.32
	Opt. (Est)	3.66	2.43	1.91	1.51	1.37	1.22	2.02
Compressor and motor weight. lb.	Cons. (up)	1080	1060	1060	1000	1000	-	1050
	Cons. (low)	400	300	300	250	280	-	300
	Opt. (Est)	270	200	200	165	165	-	200
Compressor, motor and liquefier weight lbs.	Cons. (up)	1210	1400	1180	1350	1330	-	1400
	Cons. (low)	530	650	430	600	580	760	650
	Opt. (Est)	350	440	290	400	390	150	550
Radiator weight, lbs.	Cons.	131	87	69	54	56	46	75
	Opt.	120	79	62	49	45	40	66
Total weight of compressor, motor liquefier and radiator, lbs.	Cons. (up)	1341	1487	1249	1404	1386	-	1475
	Cons. (low)	650	729	492	649	625	806	716
	Opt. (Est)	470	519	352	449	435	190	616
Power generation subsystem weight. lbs.	Cons.	440	350	280	230	240	195	300
	Opt.	295	230	180	150	140		210
Radiator area. sq. ft.	Cons.	90.5	60	47.5	37.5	38.5	31.5	52.0
	Opt.	82.5	54.5	43	34	31	27.5	45.5
Compressor Volume. cu. ft.	Cons. (up)	NO DATA AVAILABLE					-	
	Cons. (low)						-	
	Opt. (Est)						-	
Compressor and liquefier volume. cu. ft.	Cons. (up)	NO DATA AVAILABLE						
	Cons. (low)						1.6	
	Opt. (Est)						1	

TABLE 42

RELIQUEFACTION OF F_2 AT 100 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
Coeff. of performance	Cons. Opt. (Est)	0.0307 0.0338	0.0460 0.0506	0.0585 0.0643	0.0742 0.0816	0.069 0.086	0.088 0.101	0.050 0.058
Required input power. kW for 1000 lb/day	Cons. Opt. (Est)	2.93 2.67	1.96 1.78	1.54 1.40	1.21 1.10	1.30 1.05	1.02 0.89	1.8 1.55
Compressor and motor weight. lb.	Cons. (up) Cons. (low) Opt. (Est)	1000 300 200	1000 250 165	1000 250 165	1000 250 165	1000 250 165	- - -	1000 250 165
Compressor, motor and liquefier weight lbs.	Cons. (up) Cons. (low) Opt. (Est)	1100 400 270	1330 580 390	1100 350 230	1330 580 390	1330 580 390	- 730 140	1330 580 390
Radiator weight. lbs.	Cons. Opt.	95 87	64 58	50 46	39 36	43 34	33 29	59 51
Total weight of compressor, motor liquefier and radiator. lbs.	Cons. (up) Cons. (low) Opt. (Est)	1195 487 357	1394 638 448	1150 396 276	1369 616 426	1373 614 424	- 763 169	1575 640 450
Power generation subsystem weight. lbs.	Cons. Opt.	350 240	270 180	220 140	175 120	190 115	160 116	245 160
Radiator area. sq. ft.	Cons. Opt.	66 60	44 40	34.5 31.5	27 25	29.5 23.5	23 20	40.5 35
Compressor Volume. cu. ft.	Cons. (up) Cons. (low) Opt. (Est)	NO DATA AVAILABLE					- - -	
Compressor and liquefier volume. cu. ft.	Cons. (up) Cons. (low) Opt. (Est)	NO DATA AVAILABLE					- 1.2 0.8	

TABLE 43

RELIQUEFACTION OF O_2 AT 100 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
Coeff. of performance	Cons. Opt. (Est)	0.0330 0.0363	0.0495 0.0545	0.0630 0.0693	0.0798 0.0878	0.073 0.091	0.093 0.107	0.0525 0.060
Required input power. kW for 1000 lb/day.	Cons. Opt. (Est)	3.40 3.09	2.26 2.06	1.78 1.62	1.40 1.28	1.53 1.23	1.21 1.05	2.14 1.87
Compressor and motor weight. lb.	Cons. (up) Cons. (low) Opt. (Est)	1050 350 230	1050 300 200	1000 250 165	950 250 165	1000 250 165	- - -	1050 250 165
Compressor, motor and liquefier weight. lbs.	Cons. (up) Cons. (low) Opt. (Est)	1170 470 310	1400 650 430	1120 370 250	1300 600 400	1330 580 400	- 760 140	1400 600 400
Radiator weight. lbs.	Cons. Opt.	111 101	74 67	58 53	46 42	50 40	39 34	69 61
Total weight of compressor, motor liquefier and radiator, lbs.	Cons. (up) Cons. (low) Opt. (Est)	1281 571 411	1474 717 497	1178 423 303	1346 642 442	1380 620 440	- 799 174	1470 661 461
Power generation subsystem weight. lbs.	Cons. Opt.	405 260	305 200	250 160	195 135	215 125	175 115	285 180
Radiator area. sq. ft.	Cons. Opt.	76.5 69.5	51 46.5	40. 36.5	31.5 29	34.5 27.5	27 23.5	48 42
Compressor Volume. cu. ft.	Cons. (up) Cons. (low) Opt. (Est)	NO DATA AVAILABLE					- - -	
Compressor and liquefier volume. cu. ft.	Cons. (up) Cons. (low) Opt. (Est)	NO DATA AVAILABLE					- 1.6 0.9	

TABLE 44

LIQUEFACTION OF N_2 AT 1000 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He	Claude Cycle, Single Expansion
kW-hr/lb.	Cons.	1.87	1.26	0.98	0.78	0.84	0.65	0.80	0.69
	Opt. (Est)	1.68	1.13	0.88	0.70	0.63	0.55	0.68	0.59
Required input power. kW for 1000 lb/day.	Cons.	78	52.5	41	32.5	35	27	33.5	28.5
	Opt. (Est)	70	47	36.5	29	26.5	23	28.5	24.5
Compressor and motor weight. lb.	Cons. (up)	5300	3900	3400	2950	3100	-	3000	2650
	Cons. (low)	4000	2900	2300	1850	1700	-	1850	1600
	Opt. (Est)	2670	1930	1530	1230	1130	-	1230	1070
Compressor, motor and liquefier weight lbs.	Cons. (up)	6630	5200	4730	4250	4130	-	4000	3530
	Cons. (low)	5330	4200	3630	3150	2730	2470	2850	2480
	Opt. (Est)	3550	2800	2420	2110	1820	1200	1890	1650
Radiator weight. lbs.	Cons.	5100	3300	2350	1800	2000	1450	1800	1500
	Opt.	4500	2900	2050	1550	1550	1150	1500	1250
Total weight of compressor, motor liquefier and radiator. lbs.	Cons. (up)	11730	8500	7080	6050	6130	-	5800	5030
	Cons. (low)	9830	7100	5680	4700	4280	3920	4350	3730
	Opt. (Est)	8050	5700	4470	4160	3370	2350	3390	2900
Power generation subsystem weight. lbs.	Cons.	3250	2550	2100	1850	1900	1600	1930	1650
	Opt.	2050	1600	1350	1150	1080	950	1180	1050
Radiator area. sq. ft.	Cons.	1760	1180	920	730	790	610	755	640
	Opt.	1580	1060	820	655	650	520	640	550
Compressor Volume. cu. ft.	Cons. (up)	340	210	160	125	140	-	125	105
	Cons. (low)	240	140	100	80	65	-	75	60
	Opt. (Est)	160	93	67	53	43	-	50	40
Compressor and liquefier volume. cu. ft.	Cons. (up)	add on approximately 33% to the above volumes.					-		
	Cons. (low)						31		
	Opt. (Est)						20		

TABLE 45

LIQUEFACTION OF F_2 AT 1000 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
kW-hr/lb.	Cons.	1.26	0.84	0.66	0.52	0.59	0.38	0.57
	Opt. (Est)	1.13	0.76	0.59	0.47	0.44	0.32	0.48
Required input power. kW for 1000 lb/day.	Cons.	52.5	35.0	27.5	21.7	24.0	15.8	23.8
	Opt. (Est)	47.0	31.6	24.6	19.6	18.3	13.3	20.0
Compressor and motor weight. lb.	Cons. (up)	3900	3100	2600	2250	2500	-	2500
	Cons. (low)	2900	2000	1600	1300	1200	-	1350
	Opt. (Est)	1940	1330	1070	870	800	-	900
Compressor, motor and liquefier weight lbs.	Cons. (up)	4850	4130	3550	3280	3330	-	3330
	Cons. (low)	3850	3030	2550	2330	2030	2130	2180
	Opt. (Est)	2570	2020	1700	1560	1360	930	1450
Radiator weight. lbs.	Cons.	3250	1980	1460	1100	1270	710	1270
	Opt.	2850	1750	1270	980	890	585	965
Total weight of compressor, motor liquefier and radiator. lbs.	Cons. (up)	8100	6110	5010	4380	4600	-	4600
	Cons. (low)	6700	4730	3820	3310	2920	2840	3145
	Opt. (Est)	5420	3770	2970	2540	2250	1515	2415
Power generation subsystem weight. lbs.	Cons.	2700	1930	1700	1350	1500	1070	1450
	Opt.	1750	1280	1050	910	880	700	920
Radiator area. sq. ft.	Cons.	1180	790	620	490	555	356	535
	Opt.	1060	715	555	442	412	300	450
Compressor Volume. cu. ft.	Cons. (up)	210	140	100	80	87	-	85
	Cons. (low)	140	80	60	45	42	-	45
	Opt. (Est)	94	53	40	30	28	-	30
Compressor and liquefier volume. cu. ft.	Cons. (up)	add on approximately 33% to above volumes.					-	
	Cons. (low)						18	
	Opt. (Est)						12	

TABLE 46

LIQUEFACTION OF O_2 AT 1000 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
kW-hr/lb.	Cons. Opt. (Est)	1.38 1.24	0.92 0.82	0.72 0.65	0.57 0.51	0.66 0.49	0.38 0.32	0.63 0.53
Required input power. kW for 1000 lb/day.	Cons. Opt. (Est)	57.5 51.5	38.4 34.2	30.0 27.1	23.8 21.3	27.5 20.4	15.8 13.3	26.2 22.1
Compressor and motor weight. lb.	Cons. (up) Cons. (low) Opt. (Est)	4400 3050 2040	3300 2200 1470	2750 1750 1170	2500 1450 970	2600 1350 900	- - -	2500 1450 970
Compressor, motor and liquefier weight lbs.	Cons. (up) Cons. (low) Opt. (Est)	5420 4070 2720	4400 3300 2200	3770 2790 1860	3600 2550 1700	3470 2220 1480	- 2130 950	3330 2280 1520
Radiator weight. lbs.	Cons. Opt.	3750 3250	2180 1960	1550 1450	1270 1090	1460 975	710 585	1355 1110
Total weight of compressor, motor liquefier and radiator. lbs.	Cons. (up) Cons. (low) Opt. (Est)	9170 7320 5970	6580 5260 4160	5320 4240 3310	4870 3640 2790	4930 3195 2455	- 2840 1515	4685 3390 2630
Power generation subsystem weight. lbs.	Cons. Opt.	2900 1800	2020 1300	1700 1130	1450 960	1700 930	1070 700	1550 990
Radiator area. sq. ft.	Cons. Opt.	1290 1160	865 770	675 610	535 480	620 460	356 300	590 495
Compressor Volume. cu. ft.	Cons. (up) Cons. (low) Opt. (Est)	240 165 110	150 95 63	110 70 47	85 50 33	100 45 30	- - -	85 50 33
Compressor and liquefier volume. cu. ft.	Cons. (up) Cons. (low) Opt. (Est)	add on approximately 33% to above volumes					- 18 12	

TABLE 47

LIQUEFACTION OF N_2 AT 100 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
kW-hr/lb.	Cons.	2.10	1.41	1.11	0.87	0.88	0.73	0.98
	Opt. (Est)	1.90	1.27	1.00	0.78	0.66	0.62	0.83
Required input power kW for 100 lb/day.	Cons.	8.75	5.85	4.62	3.62	3.67	3.04	4.08
	Opt. (Est)	7.90	5.30	4.16	3.24	2.75	2.58	3.46
Compressor and motor weight. lb.	Cons. (up)	1500	1300	1200	1150	1150	-	1150
	Cons. (low)	660	430	400	350	300	-	400
	Opt. (Est)	440	285	265	235	200	-	265
Compressor, motor and liquefier weight lbs.	Cons. (up)	1720	1730	1420	1580	1530	-	1530
	Cons. (low)	880	860	620	780	680	950	780
	Opt. (Est)	590	570	415	520	450	270	520
Radiator weight. lbs.	Cons.	358	218	151	118	120	99	133
	Opt.	320	190	136	106	90	84	113
Total weight of compressor, motor liquefier and radiator. lbs.	Cons. (up)	2078	1948	1571	1698	1650	-	1663
	Cons (low)	1200	1050	756	886	770	1049	893
	Opt. (Est)	910	760	551	626	540	354	633
Power generation subsystem weight. lbs.	Cons.	780	580	485	410	420	360	450
	Opt.	530	410	340	280	250	240	285
Radiator area. sq. ft.	Cons.	197	132	184	81.5	82.5	68.5	92
	Opt.	178	119	94	73	62	58	78
Compressor Volume. cu. ft.	Cons. (up)	NO DATA AVAILABLE					-	
	Cons. (low)						-	
	Opt. (Est)						-	
Compressor and liquefier volume. cu. ft.	Cons. (up)	NO DATA AVAILABLE					-	
	Cons. (low)						4	
	Opt. (Est)						2	

TABLE 48

LIQUEFACTION OF F_2 AT 100 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
kW-hr/lb.	Cons.	1.42	0.95	0.75	0.59	0.63	0.50	0.69
	Opt. (Est)	1.28	0.85	0.67	0.53	0.47	0.425	0.585
Required input power. kW for 100 lb/day.	Cons.	5.90	3.96	3.12	2.46	2.62	2.08	2.87
	Opt. (Est)	5.30	3.54	2.79	2.21	1.96	1.77	2.43
Compressor and motor weight. lb.	Cons. (up)	1300	1150	1100	1050	1050	-	1100
	Cons. (low)	430	400	300	300	275	-	300
	Opt. (Est)	285	265	200	200	185	-	200
Compressor, motor and liquefier weight lbs.	Cons. (up)	1440	1530	1240	1430	1400	-	1430
	Cons. (low)	570	780	420	680	625	850	630
	Opt. (Est)	380	490	280	450	415	200	420
Radiator weight. lbs.	Cons.	206	128	100	80	86	68	94
	Opt.	180	115	90	72	64	58	79
Total weight of compressor, motor liquefier and radiator. lbs.	Cons. (up)	1646	1658	1340	1510	1486	-	1524
	Cons. (low)	750	895	510	752	689	918	709
	Opt. (Est)	560	605	370	522	479	258	499
Power generation subsystem weight. lbs.	Cons.	580	430	380	330	350	275	360
	Opt.	410	290	250	210	190	180	230
Radiator area. sq. ft.	Cons.	133	89	70	55.5	59	46.5	64.5
	Opt.	119	79.5	62.5	49.5	44	40	54.5
Compressor Volume. Cu. ft.	Cons. (up)	NO DATA AVAILABLE					-	
	Cons. (low)						-	
	Opt. (Est)						-	
Compressor and liquefier volume. cu. ft.	Cons. (up)	NO DATA AVAILABLE					-	
	Cons. (low)						2.4	
	Opt. (Est)						1.6	

TABLE 49

LIQUEFACTION OF O_2 AT 100 lbs/day

System		Single Stage J-T, using N_2	Single J-T Stage Precooled to 250°K	Two Stage J-T without Precooling using N_2	Two Stage with Precooling to 250°K	Systems with One Expander using He	Stirling Cycle, Single Stage	Gifford McMahon Engine with He
KW-hr/lb.	Cons.	1.56	1.05	0.82	0.65	0.69	0.55	0.76
	Opt. (Est)	1.40	0.95	0.74	0.585	0.52	0.47	0.645
Required input power. KW for 100 lb/day.	Cons.	6.50	4.37	3.42	2.70	2.88	2.30	3.17
	Opt. (Est)	5.85	3.96	3.08	2.43	2.17	1.96	2.58
Compressor and motor weight. lb.	Cons. (up)	1350	1175	1100	1050	1100	-	1100
	Cons. (low)	500	400	350	300	300	-	300
	Opt. (Est)	335	265	230	200	200	-	200
Compressor motor and liquefier weight. lbs.	Cons. (up)	1520	1565	1270	1440	1470	-	1470
	Cons. (low)	670	790	520	690	670	900	670
	Opt. (Est)	450	530	350	460	450	200	450
Radiator weight. lbs	Cons.	250	142	112	88	94	75	103
	Opt.	210	130	100	80	70	64	87
Total weight of compressor, motor liquefier and radiator. lbs.	Cons. (up)	1770	1707	1382	1528	1564	-	1573
	Cons. (low)	880	920	620	770	740	975	757
	Opt. (Est)	660	660	460	540	520	264	537
Power generation subsystem weight. lbs.	Cons.	640	470	405	350	360	305	380
	Opt.	450	315	260	230	210	190	240
Radiator area. sq. ft.	Cons.	146	98	77	61	65	52	71
	Opt.	131	89	69.5	55	49	44	60.5
Compressor volume. cu. ft.	Cons. (up)	NO DATA AVAILABLE					-	
	Cons. (low)						-	
	Opt. (Est)						-	
Compressor and liquefier volume. cu. ft.	Cons. (up)	NO DATA AVAILABLE					-	
	Cons. (low)						2.7	
	Opt. (Est)						1.6	

TABLE 50

RELIQUEFACTION OF H_2 AT 1000 (100) lbs/day

System		Single J-T with Liquid N_2 Precooling at 65°K		Systems with One Expander.		Compound Systems with H_2 J-T.		Stirling Cycle with H_2 J-T.	
Coeff. of performance	Cons.	0.012	(0.012)	0.015	(0.015)	0.019	-	-	(0.012)
	Opt. (Est)	0.015	(0.015)	0.019	(0.019)	0.022	-	-	(0.015)
Required input power. kW for 1000 lb/day. (100)	Cons.	197	(19.7)	158	(15.8)	125	-	-	(19.7)
	Opt. (Est)	158	(15.8)	125	(12.5)	108	-	-	(15.8)
Compressor and motor weight. lb.	Cons. (up)	8000	(2600)	7300	(1900)	6400	-	-	(1700)
	Cons. (low)	6900	(1050)	6300	(850)	5500	-	-	(750)
Compressor, motor and liquefier weight lbs.	Cons. (up)	10300	2950	9700	(2500)	8500	-	-	(2270)
	Cons. (low)	9200	(1400)	8400	(1130)	7300	-	-	(1000)
		6100	(930)	5600	(760)	4900	-	-	(670)
Radiator weight. lbs.	Cons.	15500	(980)	12000	(710)	9200	-	-	(980)
	Opt.	12000	(710)	9200	(530)	7700	-	-	(710)
Total weight of compressor, motor liquefier and radiator lbs.	Cons. (up)	26200	(4480)	21700	(3210)	17700	-	-	(3250)
	Cons. (low)	21200	(2110)	17600	(1660)	15000	-	-	(1710)
	Opt. (Est)	18100	(1640)	14800	(1290)	12600	-	-	(1380)
Power generation subsystem weight. lbs.	Cons.	5700	(1300)	4950	(1100)	4200	-	-	(1300)
	Opt.	3000	(800)	2500	(650)	2350	-	-	(800)
Radiator area. sq. ft.	Cons.	4450	(445)	3550	(355)	2800	-	-	(445)
	Opt.	3550	(355)	2800	(280)	2450	-	-	(355)
Compressor Volume. cu. ft.	Cons. (up)	1000	(70)	850	50	650	-	-	(23)
	Cons. (low)	750	(30)	550	20	450	-	-	(14)
Compressor and liquefier volume. cu. ft.	Cons. (up)	NO DATA AVAILABLE							
	Cons. (low)								
	Opt. (Est)								

TABLE 51

RELIQUEFACTION OF He AT 1000 (100) lbs/day

System		He J-T with H_2 J-T Pre-cooling at 21°K	Compound Systems with Expanders and He J-T	Stirling cycle with He J-T
Coeff. of performance	Cons. Opt. (Est)	0.0013 0.00165	0.0017 0.0020	(0.0009) (0.0012)
Required input power. kW for 1000 lb/day.	Cons. Opt. (Est)	82 65	63 53	(12) (8.9)
Compressor and motor weight. lb.	Cons. (up) Cons. (low)	5300 3800	3750 3000	(1450) (550)
Compressor, motor and liquefier weight lbs.	Cons. (up) Cons. (low) Opt. (Est)	7100 5100 3400	5000 4000 2700	(1930) (730) (490)
Radiator weight. lbs.	Cons. Opt.	5550 4100	3950 3200	(510) (360)
Total weight of compressor, motor liquefier and radiator lbs.	Cons. (up) Cons. (low) Opt. (Est)	12650 9200 7500	8950 7200 5900	(2440) (1090) (850)
Power generation subsystem weight lbs.	Cons. Opt.	3600 2100	3000 1750	(900) (550)
Radiator area. sq. ft.	Cons. Opt.	1840 1460	1420 1190	(270) (200)
Compressor Volume cu. ft.	Cons. (up) Cons. (low)	360 220	270 170	(14) (8)
Compressor and liquefier volume. cu. ft.	Cons. (up) Cons. (low) Opt. (Est)	NO DATA AVAILABLE		

TABLE 52

LIQUEFACTION OF H_2 AND He.

The typical data below are taken from text. For performances of other cycles, see Section V.

Hydrogen liquefaction Cycles	kW-hr/lb.	Total input Power (kW) for 1000 lbs/day
Single J-T with liquid N_2 precooling at 65°K	≥11.05	≥460
Single J-T system using neon	≥10.0	≥417
Dual pressure cycle with one expander and liquid N_2 precooling at 65°K	≥ 9.8	≥405
Claude cycle with one expander and liquid N_2 precooling at 65°K	≥ 8.75	≥366
<u>Helium liquefaction</u>		
Single J-T cycle with 21°K precooling obtained by H_2 J-T system with liquid N_2 precooling at 65°K	≥ 9.8	≥410
Compound system with three expanders and J-T He liquefaction	≥ 8.85	≥370

TABLE 53

PARTIAL LISTING OF MINIATURE CLOSED-CYCLE REFRIGERATION SYSTEMS FOR COOLING TO 77°K

System	Refrig. Temp. °K	Refrig. Load Watts	Cooldown Time min.	Total Input Power Watts	System Weight lbs.	System Volume cu. ft.	C.P.
Hughes Aircraft Co.* Single J-T using N_2	77	12	5	750	40	-	0.016
Garrett Corp. Single J-T.**	77	1	-	130	20		0.0078
Santa Barbara Res. Center.*** Single J-T Model No. 79K-20W.	79	5	5	326	16	0.35	0.015
Air Products and Chemicals Inc.# Single J-T using N_2	80	1-5	-	300-1000	16-35	0.37	0.0033-0.005
Cryogenerators. Division North Amer. Philips, Co. Inc.## Cryogem. Stirling cycle.	77	9 5	< 7	180	12 (less motor)	-	0.053
Malaker Laboratories Inc.### Cryomite Mark VII Stirling cycle.	77	9	6	170	11	0.15	0.053

Notes for this table are located on page 123.

VIII.3 SOME COMPARATIVE DATA ON MINIATURE SYSTEMS

The data on the performance, weight, volume, etc., of the cycles considered in this report for reliquefaction or liquefaction of gases has been gathered and presented for systems of small to medium size such as would allow for liquefaction up to 1000 lb/day. It is very difficult to extrapolate the data presented with any accuracy towards the origin, i.e. for systems providing only about 1 to 5 watts of refrigeration. In order to make some survey of this region of miniature refrigeration systems and to provide some comparison of cycles in this region, we considered the most significant approach lies in enumeration of the properties and operation data of some existing miniature refrigerators. Table 53 gives a partial listing of known miniature closed-cycle refrigerators for operation down to 77°K, giving, where known, published data on their performances, (C.P.), weight and volume. Table 54 gives a similar partial listing for miniature closed-cycle refrigeration systems for cooling below 77°K. It must be pointed out that these tabulations are not intended to be exhaustive, but rather to serve as an indication of the relative performance of some of the major closed cycles.

Remarks on the relative merits of the different miniature systems for space operation are made in Subsection VIII.4 in which the systems of all sizes covered by this report are discussed. It is to be noted that the general comparisons made in Subsection VIII.1 are equally applicable to miniature as well as to larger systems.

NOTES FOR TABLE 53

*Hughes Aircraft Co. Brochure 2784.20/46. 14 Jan. 1963
**Garret Corp. AiReasearch Div. Report No. AE-2088R. Sept. 27 1962
***Santa Barbara Res. Center. Brochure dated 12/3/62.
#Air Products and Chemicals Inc. Brochure "Advanced Products Data Sheet". (undated).
##Cryogenerators. Division North American Philips Co. Inc. Norelco Reporter Vol. IX. No. 3 May-June 1962 and Brochure CP1004-10M-V6/63 and paper by DuPre, F. IRIS Detector Specialty Group Meeting. Syracuse. 17 June 1963. (Input power based on Author's assumption of 70% motor efficiency).
###Malaker Laboratories Inc Brochure "The Cryomite and private information. (Motor efficiency 70%)

TABLE 54

PARTIAL LISTING OF MINIATURE CLOSED-CYCLE
REFRIGERATION SYSTEMS FOR COOLING BELOW 77°K

System	Refrig. Temp. °K	Refrig. Load Watts	Cooldown Time min.	Total Input Power Watts	System Weight lbs	System Volume cu. ft.	C. P.
Hughes Aircraft Co.* Single stage Solvay.	30	2	10	575	25	-	0.0035
Cryogenerators Division North Amer. Philips Co. Inc.** Cryogen. Stirling cycle.	30	1.5	10	265	12 (less motor)	-	0.0057
Malaker Laboratories Inc.*** Cryomite Mark VI-C Single-stage Stirling cycle.	30	2.5	15	290	11	0.15	0.0086
Garrett Corp.# Claude bypass system using Ne.	25	2	-	750	30	0.6	0.0027
Hughes Aircraft Co.* Two stage Solvay.	12	0.4	25	575	approx. 30	-	0.0007
A.D. Little Inc.## Three-stage Gifford-McMahon engine with He J-T.	4.2 4.2	0.75 4	- -	2250 7500	273 850	3.8 24	0.00033 0.00053
Hughes Aircraft Co.* Three-stage Solvay with He J-T.	4.2	2	60	5000	-	-	0.0004

TABLE 54 (Cont'd)

PARTIAL LISTING OF MINIATURE CLOSED-CYCLE
REFRIGERATION SYSTEMS FOR COOLING BELOW 77°K

System	Refrig. Temp. °K	Refrig. Load Watts	Cooldown Time min.	Total Input Power Watts	System Weight lbs	System Volume cu. ft.	C. P.
Air Products and Chemicals Inc.### Three-stage J-T.	4.4	1.25	10 hours	7500	-	-	0.00017
Garrett Corp# Claude system using Ne and He.	4.2	1	-	4950	178	8.0	0.00020
Air Products and Chemicals Inc.*# Two expanders and He J-T.	2.5	0.2	3 hours	-	150	4	-

*Hughes Aircraft Co. Brochure 2784.20/46. 14 Jan 1963
**Cryogenerators. Division North American Philips Co. Inc. Norelco Reporter Vol. IX. No. 3. May-June 1962 and Brochure CP1004-10M-U6/63 and paper by DuPre, F. IRIS Detector Specialty Group Meeting. Syracuse 17 June 1963 (Input power based on author's assumption of 75% motor efficiency.)
***Malaker Laboratories Inc. Private information. (Motor efficiency 75%.)
#Garrett Corp. AiResearch Division. Report No. AE-2088R. Sept. 1962.
##See McMahon, H. O. Cryogenics. 2. 65. 1960 and A. D. Little Inc. Brochures on "The Cryodyne".
###See Lashmet, P. K. and Geist, J. M. Proc. Cryo. Eng. Conf. 1962.
*#See Zeitz, K. and Woolfenden, B. K. Proc. Cryo. Eng. Conf. 1962.

VIII.4 DISCUSSION OF THE QUALITATIVE AND QUANTITATIVE COMPARISON OF CYCLES AND CONCLUSIONS

4.1 RELATIVE POWER REQUIREMENTS FOR LIQUEFACTION AND RELIQUEFACTION

The relative efficiencies of liquefaction and reliquefaction have been discussed theoretically for ideal reversible cycles in Section IV. The results of this discussion were summarized in Tables 4, 5, 6. In Table 6 the theoretical evaluations of the ratio, r, of the minimum work for liquefaction to the minimum work for reliquefaction were given for various gases, assuming the liquid to be produced at a pressure of 1 atm. and assuming a sink temperature of 293°K.

By inspection of Tables 1 through 52 an assessment can be made of this ratio, r, for practical liquefiers and reliquefiers for which allowance has been made for all losses and irreversibilities. The results of this assessment are given in Table 55, together with the previously presented data on r for ideal reversible cycles.

TABLE 55

RATIO, r, OF THE MINIMUM WORK FOR LIQUEFACTION TO MINIMUM WORK FOR RELIQUEFACTION IN THEORY AND IN PRACTICE.

(Pressure of liquid 1 atm Sink temperature 293°K)

Substance	r (Theory)	r(Practical)
Oxygen	1.29	approx. 2
Nitrogen	1.33	approx. 2
Hydrogen	1.88	approx. 2.5 to 3
Helium	4.75	approx. 6

4.2 GOOD CYCLES FROM POWER REQUIREMENT VIEWPOINT

It is convenient to discuss the question of the overall power requirements for practical cycles under two headings, namely (a) cycles for operation down to 77°K and (b) cycles for liquefaction or reliquefaction of H_2 and He.

(a) Data for the coefficient of performance for reliquefiers and for the number of kW-hr/lb. for liquefaction for N_2, O_2 and F_2 are given in Tables 38 through 49 for liquid production

rates of 1000 and 100 lbs/day. Moreover graphs and tabulations have been presented in Section V giving fuller data on the same parameters over the whole range of liquid production covered in this report. Reference is made to the above mentioned graphs and tables and to Table 53, which gives some data on miniature systems. The four best cycles of those considered from the point of view of minimum practical overall power requirements are: (1) Two-stage J-T with precooling, (2) systems with one expander, (3) Stirling cycle, (4) Gifford-McMahon engine and, for liquefaction of N_2, (5) the Claude cycle with single expansion. Mention should be made at this time also of the cascaded compressed vapor systems which, as has been pointed out in Section V.1, can be made to have very high practical efficiency. However for reasons which have been discussed in Section VIII.1 and tabulated in Table 36, the cascaded compressed vapor system, due to the large number of components involved, is not considered suitable for long period space operation and is not further considered.

In referring to the performance data tabulated in Tables 38 through 49 and in Table 53 the difficulties in assessing the probable error in or ranges of the evaluations should be re-emphasized here. Again it must be pointed out that the performance data must not be interpreted as necessarily representing the best that can be achieved by any particular cycle.

It is clear from the data referred to above that the J-T systems, with the exception of the two-stage precooled J-T system, have markedly lower coefficients of performance as reliquefiers and require considerably higher kW-hr/lb liquefied than the cycles (2) through (5) enumerated above. Basically this is due to the inherent irreversibilities in all J-T cycles, as discussed in Section VIII.1.1, and to the basic dependence of such cycles on the non-ideality of the working fluid. As will be noted in the subsections following, J-T systems possess other undesirable features for long period space operation besides their relatively inefficient performance noted herewith, and probably therefore should not be further considered in detail. However an important point of a general nature should be made at this juncture. As is clearly evident from Tables 38 through 51 and as will be further clarified in the discussion of Figs. 78, 79, and 80, any increases in the overall power requirements for reliquefiers or liquefiers are accompanied by approximately proportional increases in the system weight and volume and in the radiator area (and weight) requirements. The issue of efficiency of the cycle, therefore, is a paramount one and any compromise in efficiency can only be justified if there are overwhelming and overriding disadvantages inherent in an efficient cycle with regard to reliability or adaptability to space environment.

In comparing the better cycles, (1) through (5) listed above, it will be seen that they are all roughly of the comparable efficiency, at least for operation at 77°K. The Stirling cycle appears to have an edge on the others and this superiority is more clearly evident at the higher temperatures for reliquefaction or liquefaction of F_2 and O_2. This fact probably reflects the intrinsic high theoretical efficiency of the Stirling cycle, which is discussed in Section V, and which is evident in Table 37. It is worth noting in this connection that the Stirling cycle does not suffer significantly from the intrinsic drawback of exchange of latent heat for sensible heat, which is characteristic of systems employing expansion engines only. The high efficiency of the Stirling cycles and Gifford-McMahon engines in this temperature range is also significantly due to the fact that they employ regenerators rather than heat interchangers, since for this temperature range highly efficient regenerators of small volume can be easily constructed whereas this is not so in the case of heat exchangers.

In choosing a reliquefier or liquefier for N_2, F_2 or O_2 therefore the choice should be made from the systems (1) through (5), enumerated above and because of their comparable efficiencies, the final choice must be determined only after consideration of the other factors involved in their operation in space vehicles, and after consideration of the required rates of liquefaction (reliquefaction).

However before proceeding to these other considerations it is of value to consider the question of the applicability of two distinctly preferable cycles for operation down to 77°K in terms of the desired rates of liquefaction (reliquefaction). The two distinctly preferable cycles are (2) Systems with one expander, the expander being a turbine (with centrifugal compressor) and (3) the Stirling cycle. The main reasons for their desirability, besides their small power requirements, weights, and volumes (and associated small areas and weights of the radiator to be used with them), lie in their absence of valves, which are a source of reliability problems, the fact that they are single phase systems and their relatively small number of moving parts and components, a factor which leads to inherent reliability. At the high production rates, up to 1000 lbs/day N_2, F_2, or O_2, both systems are highly recommended. For the lower production rates, down to miniature systems of 1 watt refrigeration capacity, difficulties may arise in turbine systems, due to the extremely small size of the turbines which must be used. Such miniature turbines could be a source of blockage by contaminants carried in the working gas and would impose very rigid requirements on the working fluid purity and on the compressor design. Moreover, even for turbines of the highest efficiency, the limiting problems on overall efficiency would continue to lie in high efficiency heat exchanger design and in the inherent disadvantage of exchange

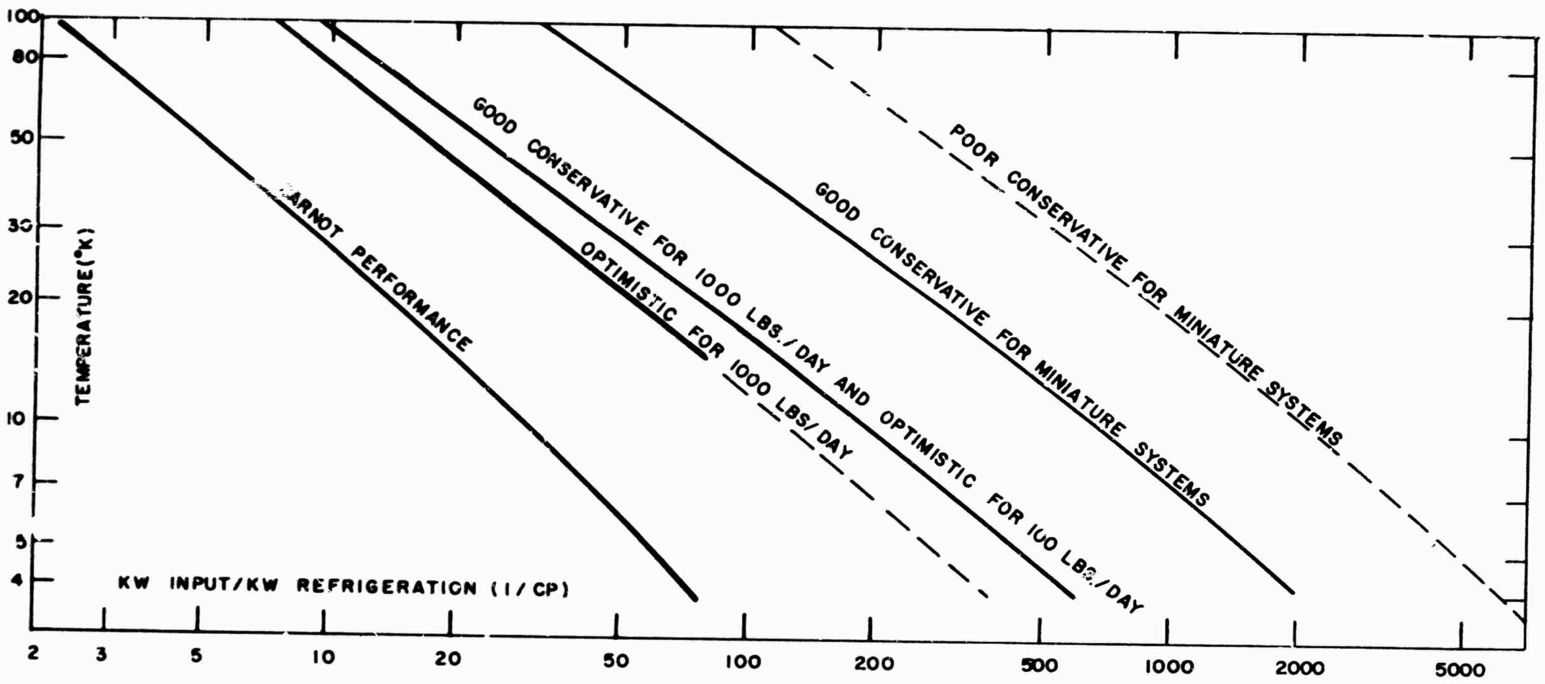

Fig. 78 - Required total input power (kW) per kW of refrigeration as a function of temperature for refrigerating systems.

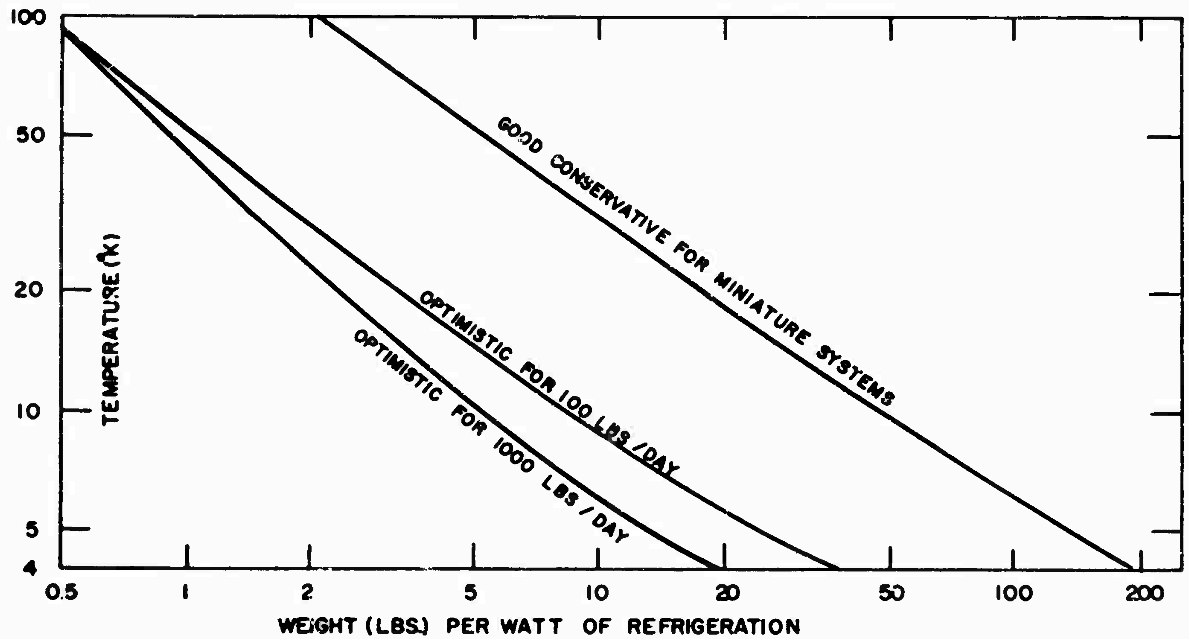

Fig. 79 - Weight of refrigeration systems per watt of refrigeration as a function of temperature.

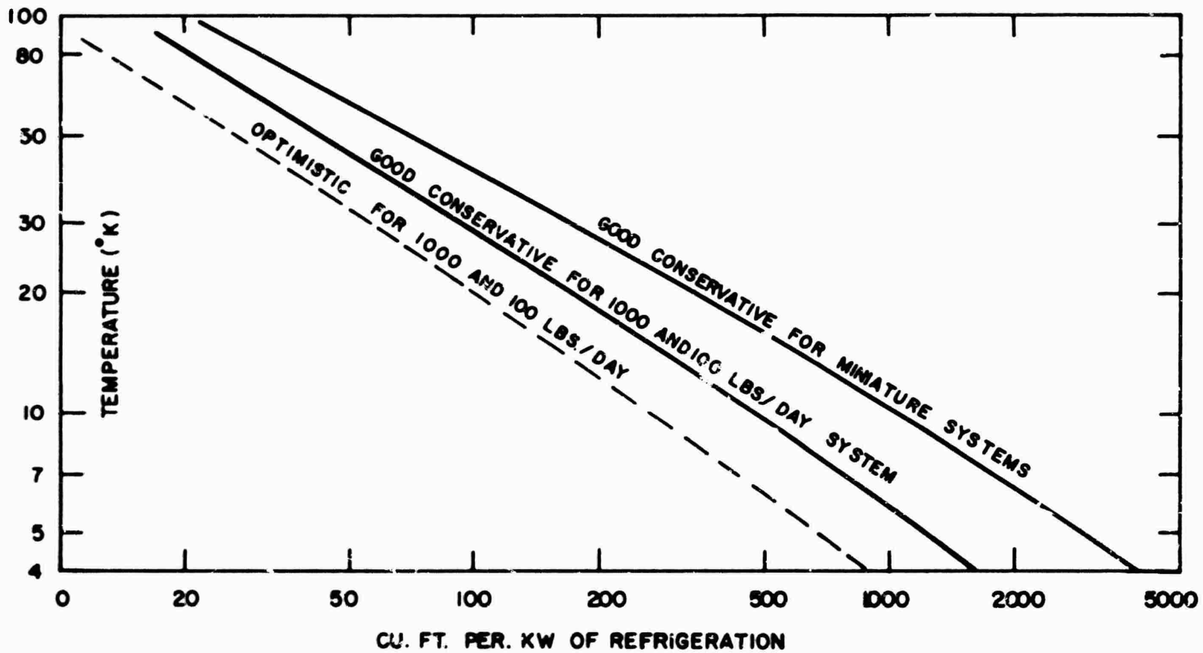

Fig. 80 - Estimates of best volumes of refrigerators per kW of refrigeration as a function of temperature.

of latent heat for sensible heat. In consequence, for the smaller scale operation, the Stirling cycle appears to be a preferred cycle.

(b) Data for the coefficients of performance for reliquefiers and for the number of kW-hr/lb for liquefaction for H_2 and He are given in Tables 50 through 52 for liquid production rates of 1000 and 100 lbs/day. Moreover detailed discussions of these factors have been given in Section V. Again, as noted earlier, the difficulties in assessing the probable error of the evaluations should be re-emphasized and again it must be pointed out that the performance data must not be interpreted as necessarily representing the best that can be achieved by any particular cycle. Of the cycles listed in the above mentioned tables, those to be preferred from a point of view of efficiency, and hence from the point of view of weight and volume also, are systems with one expander and compound systems using expanders. Again their superiority over systems using J-T expansions only is mainly to their inherently smaller thermodynamic irreversibilities. Unfortunately, due to lack of practical data on the larger production sizes, comparisons with possible compound systems using Stirling cycles cannot be made at this time, although some general discussion of such systems is given in Section V. On the other hand, because of the advantageous position of Stirling cycles in the higher temperature range (77°K to 90°K), such cycles, either as single stage or multiple stage, should be thoroughly investigated in practice for use in H_2 or He reliquefaction or liquefaction, at least in sizes up to about 50 kW total input power. For larger sizes, as has been previously discussed, systems using turbine expanders are to be preferred. For miniature sizes, as is evident from Table 54, Stirling based cycles for very low temperature operation are most suitable.

It is sometimes pointed out that for the lower temperatures of operation, below about 20°K, regenerator efficiencies fall off seriously and systems using heat exchangers are therefore to be preferred for such very low temperature operation. It appears likely, however, that a breakthrough may be expected in the development of highly efficient regenerators for very low temperature use, and this may result in highly efficient H_2 and He systems in the period 1965-1970.

Whereas for liquefaction or reliquefaction of H_2 it is possible to construct cycles which do not rely on use of a J-T expansion cycle somewhere in the system (c.f. the tables cited and Section V), this is not possible at the present state of the art for He liquefaction or reliquefaction. It would be of value to consider development of, say, turbine expanders and of more efficient regenerators used in conjunction with Stirling

based cycles, in order to obviate the necessity of this J-T circuit in He temperature systems, thus greatly reducing complexity and the number of required components with resulting gain in reliability and suitability for space operation. It is considered that this is a feasible goal for the 1965-1970 period.

To summarize briefly some of the data presented in this report and discussed above on cycle efficiencies, plots have been made of the total overall required input power in kW per kW of refrigeration (1/CP) as a function of refrigeration temperature for the average "better" cycles operating with a sink temperature of 300°K. The plots are reproduced in Fig. 78 and are for: "Carnot performance", "Optimistic for 1000 lbs/day"; "Good conservative for 1000 lbs/day and optimistic for 100 lbs/day" and "Good conservative for miniature systems." These plots show clearly the rapidly increasing power requirements for all systems, including the ideal Carnot engine, as the temperature of refrigeration is reduced. They show, however, that although the power requirements of practical systems are strongly dependent on the design refrigeration load, the variation with temperature is similar for all sizes of system. It is anticipated that these plots may be of value in refrigeration specification. In the temperature range 90°K to 20°K the curves of Fig. 78 are approximately linear and can be approximated by the following formula:

$$(1/CP) = AT^{-n} \tag{59}$$

where A and n have the values given in Table 56.

In order to emphasize the rapid increase in the power requirements of any refrigerator which occurs on decreasing the desired temperature of refrigeration, Table 57 has been drawn up which lists the number of watts of total overall input power required per watt of refrigeration as a function of temperature for the average "better" cycles of various sizes considered in this report.

4.3 GOOD CYCLES FROM WEIGHT AND VOLUME VIEWPOINT

A detailed account has been given in section VIII.2 of the way in which the weights and volume or compressor-motors, liquefiers (reliquefiers), radiators and power generation subsystems were arrived at. Again it must be noted that the evaluations of these parameters presented in Tables 38 through 51 are to be taken as rough guides only, because of the paucity of firm data on which to base the evaluations. As discussed in Section VIII 4.2 above, and as is evident from the tables cited, there is an approximate one to one relation between the total overall power requirement of a system and the weights and volumes of the components. Those cycles which have been cited in Section VIII 4.2

TABLE 56

DATA ON REFRIGERATORS (SINK TEMPERATURE 300°K)

In the range 90°K to 20°K, the reciprocal of the coefficient of performance (1/CP), the weight of the system (lbs/watt of refrigeration) and the volume (cu. ft/kW of refrigeration) can be approximated by the formula: $x = A\ T^{-n}$ where A and n have the values given below.

System	1/CP		Weight lbs/watt		Volume cu. ft/kW.	
	A	n	A	n	A	n
Optimistic for 1000 lbs/day	2510	1.27	55	1.57	8700	1.49
Optimistic for 100 lbs/day	4730	1.35	136	1.24	8700	1.49
Good conservative for 1000 lbs/day	4730	1.35	--	--	16800	1.53
Good conservative for 100 lbs/day	--	--	--	--	16800	1.53
Good conservative for miniature system	12000	1.27	990	1.34	63000	1.74

TABLE 57

REFRIGERATOR PERFORMANCE

The numbers give the watts of total overall input power required per watt of refrigeration at the temperature specified. (Sink temperature 300°K)

Temp.	Ideal Carnot Engine	Optimistic Figure for 1000 lbs/day	Good Conservative Figure for 1000 lbs/day and Optimistic for 100 lbs/day	Good Conservative Figure for Miniature Systems
80°K	2.75	9.5	12.5	43
30°K	9.0	33	46	173
20°K	14.0	54	78	290
10°K	29.0	125	190	690
4°K	74.0	360	580	2000

as desirable from a power requirement viewpoint therefore are also the desirable cycles from the weight and volume viewpoint.

On the basis of the assumptions used in arriving at the data presented in Tables 38 through 51, evaluations have been made of the average weight of the "better" refrigerators per watt of refrigeration. Included in this weight is that of the compressor-motor(s) and liquefier (refrigerator) only. The main assumptions were: (a) sink temperature 300°K; (b) temperature of fluid inlet to radiator 400°K; (c) temperature of fluid outlet to radiator 300°K; (d) equivalent radiative sink temperature, T_r, 250°K. (e) radiator fin effectiveness equal to unity. Fig. 79 shows graphically these evaluations of the weights per watt as a function of the temperature of refrigeration for: "Optimistic for 1000 lbs/day"; "Optimistic for 100 lbs/day" and "Good conservative for miniature systems". The rapid increase in weight as the refrigeration temperature is reduced and the fact that the curves resemble closely those for (1/CP) versus T given in Fig. 78 are to be noted. In the temperature range 90°K to 20°K the curves of Fig. 79 are approximately linear and can be approximated by the formula:

$$\text{Weight (lbs/watt)} = A\ T^{-n} \qquad (60)$$

where A and n have the values given in Table 56.

Similar plots have been made of the combined compressor-motor and refrigerator volumes per watt of refrigeration as a function of refrigeration temperature under the same basic assumptions and these are given in Fig. 80. Again the curves are similar to those for the weights and for (1/CP) and again in the temperature range 90°K to 20°K they can be approximated by the formula

$$\text{Volume (cu. ft./kW)} = A\ T^{-n} \qquad (61)$$

where A and n have the values given in Table 56.

Estimates of the relative weights of the radiator and power generation subsystem as compared with the weight of the combined compressor-motor and reliquefier (liquefier) for "better" cycles are given in Table 58. The data are taken from Tables 38 through 51 and therefore must be taken as rough guides only.

Considering first the relative power generation subsystem weight (C/A of Table 58). It will be seen that for the smaller scale systems (100 lbs/day) for N_2, F_2 and O_2 reliquefaction and liquefaction the C/A values range from 22% to 39% for the "conservative" compressor-motor plus reliquefier (liquefier) estimates and between 82% and 95% for the "optimistic"

TABLE 58

RELATIVE WEIGHTS OF RADIATOR AND POWER GENERATOR SUBSYSTEM AS
COMPARED WITH WEIGHT OF COMBINED COMPRESSOR-MOTOR AND RELIQUEFIER (LIQUEFIER)

(Data taken for better cycles)

System	Est. combined compressor-motor and reliquefier (liquefier) weight lbs		B Est. Radiator weight lbs		B/A percent		C Est. power generation subsystem weight lbs		C/A percent	
	cons.	opt.	cons.	opt.	Cons.	opt.	cons.	opt.	cons.	opt
Reliq. N_2 at 1000 lbs/day	1500	620	550	465	37	75	870	590	58	95
Reliq. F_2 at 1000 lbs/day	1280	480	310	250	24	52	720	500	56	104
Reliq. O_2 at 1000 lbs/day	1300	500	340	275	26	55	760	520	58	104
Reliq. N_2 at 100 lbs/day	760	150	46	40	6	27	195	120	26	80
Reliq. F_2 at 100 lbs/day	730	140	33	29	4.5	21	160	116	22	83
Reliq. O_2 at 100 lbs/day	760	140	39	34	5	24	175	115	23	82
Liq. N_2 at 1000 lbs/day	2470	1200	1450	1150	59	96	1600	950	65	79
Liq. F_2 at 1000 lbs/day	2130	930	710	585	33	63	1070	700	50	75
Liq. O_2 at 1000 lbs/day	2130	950	710	585	33	62	1070	700	50	74
Liq. N_2 at 100 lbs/day	950	270	99	84	10	31	360	240	38	89
Liq. F_2 at 100 lbs/day	850	200	68	58	8	29	275	180	32	90
Liq. O_2 at 100 lbs/day	900	200	75	64	8	32	305	190	34	95

TABLE 58 (Cont'd)

RELATIVE WEIGHTS OF RADIATOR AND POWER GENERATOR SUBSYSTEM AS COMPARED WITH WEIGHT OF COMBINED COMPRESSOR-MOTOR AND RELIQUEFIER (LIQUEFIER)

(Data taken for better cycles)

System	Est. combined compressor-motor and reliquefier (liquefier) weight lbs		B Est. Radiator weight lbs		B/A percent		C Est. power generation subsystem weight lbs		C/A percent	
	cons.	opt.	cons.	opt.	cons.	opt.	cons.	opt.	cons.	opt.
Reliq. H_2 at 1000 lbs/day	7300	4900	9200	7700	125	156	1200	2350	57	48
Reliq. H_2 at 100 lbs/day	1000	670	980	710	98	105	1300	800	130	119
Reliq. He at 1000 lbs/day	4000	2700	3950	3200	99	118	3000	1750	75	65
Reliq. He at 100 lbs/day	730	490	510	360	70	73	900	550	123	112

estimates of the same weights. The J/A values for the larger scale systems (1000 lbs/day) and for the H_2 and He systems are much larger ranging up to 130%. Clearly, the data on power generation subsystem weights given in Fig. 77, on which these estimates are based, indicate that the power generator subsystem is a major component contributing to the overall weight.

In considering radiator weights Table 58 shows that for the smaller scale systems (100 lbs/day) for liquefaction or reliquefaction of N_2, F_2, or O_2 the radiator weight is relatively a minor factor. For the larger scale systems (1000 lbs/day) operating down to 77°K the relative radiator weight becomes important under the assumed conditions of operation of the systems and may reach values as high as 96%. For the systems for H_2 and He reliquefaction and liquefaction the relative radiator weight becomes an even more serious factor, ranging up to 150%. Moreover for higher values of the equivalent radiative sink temperature, T_r, than the one chosen for these computations (250°K), the relative radiator weights would increase still further (See Fig. 73) and such higher values of T_r may be expected in certain space environments (See Fig. 72 and subsection VII.2.51). Clearly space vehicle radiator designs should be selected to minimize T_r, in order to reduce relative and absolute radiator weights as much as possible. A further step can be considered, namely that of increasing the fluid inlet and outlet temperatures to and from the radiator (See Fig. 73 for effects of this process). However, as has been pointed out previously, such a step involving as it does an increase in the sink temperature of the reliquefier (liquefier) system results in an increase in the power requirements. (See Fig. 46 for some results on this). Such an increase in the power requirements increases the total weight of the combined compressor-motor and liquefier (reliquefier) and of the power generation subsystem, as is evident from Tables 38 through 51. For any given system therefore a compromise must be reached between these opposing factors.

It must be repeated that all the evaluations of the total overall power requirements made and used here and elsewhere in this report have taken into account ~~the~~ the isothermal compressor efficiency and the motor and combined motor and motor drive efficiency. In nearly all cases (See Section V) these efficiencies have been taken to be approximately 67% and 80% to 85% respectively Both of these sources of inefficiency therefore when combined together result in increasing the basic power requirement of the refrigerator or liquefier by about a factor of two (2). This noteworthy increase in the power requirements approximately doubles the size and weight of the required radiator and the weight of the power generation subsystem, although it probably makes a relatively smaller difference to the weight or volume of the combined compressor-motor and liquefier

(reliquefier) system. Since radiator weight and area and power generation subsystem weight are very serious factors in the overall system (c.f. Tables 38 through 51 and Table 58), there is a clear need for future improvement of these sources of inefficiency.

Moreover, improved compressor design for space liquefiers and reliquefiers is to be sought, not only for the reason cited immediately above, but also because, by the use of good design, lightweight materials and oil-free operation, considerable weight and volume savings may be expected and the adaptability to space environment enhanced.

4.4 FURTHER OBSERVATIONS ON RELIABILITY

Some observations on the reliability factors in reliquefiers and liquefiers have already been made in subsection VIII 1.2 and some of the parameters affecting reliability and ease of maintenance have been set out comparatively for various cycles in Tables 36 and 37. Three factors should be re-emphasized here, namely: (a) minimization of the number of components and especially the number of moving parts; (b) avoidance of constrictions to passage of the working fluid at low temperatures in order to obviate blockage by contaminants and (c) avoidance where possible of moving valves, especially rapidly moving valves, such as are encountered in reciprocating compressors and in some of the "cold boxes" of the reliquefiers (liquefiers) discussed in this report.

With regard to the first item consideration should be given not only to the avoidance where possible of systems which inherently have many components, e.g. compound systems with multiple expanders, elaborate precooling systems, etc., but also to minimization of extra components such as filters, controls, etc. Of the five better cycles enumerated above in section VIII 4.2 for operation down to 77°K, Stirling cycles and systems using one turbo-expander (with centrifugal compressor) not only fulfill this desirable objective but also item (c) above. For larger scale systems (1000 lbs/day) down to 77°K, these two systems also fulfill the desired objective (b) above, whereas systems with J-T valves do not. On the other hand for smaller scale systems (100 lbs/day) and miniature systems, turbo expanders, as pointed out above, may be a seat of blockage difficulties and Stirling cycles are at present more desirable. In this connection, again as has been pointed out above, all compressors must be oil-free and extreme precautions must be taken to assure maximum purity of the working fluids in order to permit long periods of continuous reliable operation.

With regard to the systems for H_2 and He reliquefaction and liquefaction, except for systems using one (turbo) expander only (with centrifugal compressor) and two-stage Stirling cycles for H_2 production, all systems cited in Tables 50 through 52 violate one or more of the desirable items (a), (b) and (c) above. For He production, without some significant breakthrough in techniques, the question of reliability remains more serious than for the other gases.

A further observation concerning reliability is that high pressure systems are undesirable in that they give rise to higher stresses and require the use of reciprocating compressors using valves. J-T systems for all the gases considered in this report except helium require high pressures.

4.5 FURTHER OBSERVATIONS ON ADAPTABILITY TO SPACE ENVIRONMENT

Some observations on the desirable features of reliquefiers and liquefiers for space operation have been made in subsection VIII.1.4 and some of the parameters, especially the desirable feature of single phase working fluid, have been set out comparatively for various cycles in Tables 36 and 37. Further desirable features are: (a) The system must be unaffected by erosion due to solar wind, stationary gas in space, solar flares, cosmic radiation and by meteoroid damage. All of the better systems considered can be made to meet this objective except, as discussed in Section VII, the question of satisfactory design is a critical problem area. (b) The system must be unaffected by radiation damage. This question has been thoroughly discussed in section VII and it is considered that, with right choice of materials, all the gas reliquefying and liquefying cycles considered herewith can be made to meet this objective. (c) The system must operate under conditions of zero-g. This requirement makes systems not employing a two-phase working fluid the more desirable, such as systems (2), (3) and (4) enumerated in subsection VIII 4.2 (d) The system must operate in or withstand high-g accelerations and vibration and must be high-vacuum sealed to avoid loss of working fluid even in zero external pressure environments. To meet these requirements systems which are compact, having a minimum number of components and moving parts and a minimum number of piping joints should be favored. Here Stirling cycle systems and systems with one expander are clearly desirable.

4.6 ON SOME AREAS OF IGNORANCE IN RELIQUEFIER (LIQUEFIER) DESIGN SPECIFICATIONS

Although, as is evident from the content of this report, there are many areas of ignorance in the subject of reliquefaction and liquefaction of N_2, F_2, O_2, H_2, and He, a few seem worth setting out explicity in the hope that this may stimulate the generation of further knowledge. Those selected are:

(a) Exact compressor-motor weights and volumes, particularly for those with minimum weights and volumes and for those which operate oil free, whether reciprocating or centrifugal.

(b) Exact data on volumes and weights of actual reliquefying or liquefying systems in the ranges of production covered by this report, particularly those with minimum weights and volumes.

(c) Optimistic figures for radiator weights and effectiveness, suitable to withstand the meteoroid flux obtaining in space, and radiator reliability for the period 1965-1970.

(d) Firm data on average periods for continuous operation between maintenance for actual reliquefier or liquefier systems in the production range covered by this report.

4.7 ON SOME CRITICAL PROBLEM AREAS AND POSSIBLE FUTURE BREAKTHROUGHS

Many critical problem areas have been uncovered and discussed in the body of this report and it is unnecessary to discuss them further at this stage. However it may be of value to enumerate some of those critical problem areas which, in the opinion of the authors, will be important in the future development of space liquefiers and reliquefiers. They are:

(a) The need for increase in the isothermal efficiency of compressors.

(b) The need for increase in the efficiency of motors and motor drives, particularly for small scale units.

(c) The need for a wider selection of reliable lightweight oil-free compressors.

(d) The need for a reduction in space radiator weights (lbs/sq. ft. of radiator area) and for simultaneous improvement of their anti-meteoroid protection.

(e) The need for a reduction in power generation subsystem weight per kW output.

(f) The need for high efficiency regenerators for very low temperature operation ($\leq 20°K$).

(g) The need for simplification of He liquefaction, preferably without the use of J-T circuits.

In general by definition a breakthrough in technique is one which is unexpected and hence no trustworthy predictions are really possible in this area. However as mentioned earlier in this report, significant improvements may be expected in the near future in regenerator efficiencies at very low temperatures ($\leq 20°K$ which may lead to a breakthrough in Stirling based cycles for refrigeration at the lowest temperatures.

IX. RECOMMENDATIONS

After the detailed comparison of cycles for reliquefaction and liquefaction of O_2, F_2, N_2, H_2 and He in space environments given in Section VIII and after the discussion and conclusions arrived at also in Section VIII, it seems desirable to implement these conclusions with specific recommendations for the attainment of the desired goal.

In view of the general findings of this study, we recommend:

(1) That the Stirling cycle be adopted for reliquefaction and/or liquefaction of N_2, O_2 and F_2 for all production rates from miniature to systems requiring approximately 50 kW input power.

(2) That steps be taken to construct prototype oil-free Stirling cycle systems for the reliquefaction and/or liquefaction of N_2, O_2 or F_2, in various sizes in the production range mentioned in (1) above, with minimum weight and volume, adaptable to space environment and that reliability and environmental tests be carried out with these prototypes.

(3) That construction be initiated of test-model oil-free compound systems for the reliquefaction and/or liquefaction of H_2 and of He at various rates in the range specified in (1) above, which use the Stirling cycle as the primary refrigerating agent therein, with minimum weight and volume, adaptable to space environment and that reliability and environmental tests be carried out with these tests models.

(4) That compact oil-free systems with one expander only, in which the expander and the compressor are of rotary type, be adopted for the reliquefaction and/or liquefaction of N_2, O_2, F_2 and H_2 for the larger production rates involving input powers in excess of approximately 25 kW.

(5) That steps be taken to construct prototype compact oil-free systems with one turbo-expander (and centrifugal compressor) for the reliquefaction of N_2, O_2, F_2 or H_2 in the size range between 25 kW and 250 kW input power, with minimum weight and volume, adaptable to space environment and that reliability and environmental tests be carried out with such prototypes.

(6) That construction be initiated of test-model compact oil-free compound systems for the reliquefaction and liquefaction of He at the highest rate considered in this report which use one turbo-expander only as the primary refrigerating agent and which use centrifugal compressors only, with minimum weight and volume, adaptable to space environment and that reliability and environmental tests be carried out with such test models.

(7) That efforts be made to stimulate the development of reliable compressors, particularly centrifugal compressors, in the power range covered in this report which shall be oil-free, of minimum weight, and volume and which development shall put major emphasis on attaining highest isothermal efficiency and highest combined motor and motor-drive efficiency.

(8) That improvements be made in space radiator design to minimize weight per watt radiated, and to maximize resistance to meteoroid damage and to be compatible with space reliquefier (liquefier) requirements.

(9) That development be encouraged in the design and construction of helium reliquefiers (liquefiers) which shall be the simplest character obviating the requirement for a J-T circuit.